MARINE PROPELLERS

SYDNEY W. BARNABY

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BY

SYDNEY W.alker BARNABY,

M. INST. C.E. AND M.I.N.A.

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PREFACE

TO THIRD EDITION.

In issuing a third edition of this book it appeared advisable to reconstruct and enlarge it in order to introduce new material which had become available. The original form of lectures as delivered to a class of students has been abandoned and the book rewritten and brought up to date.

No attempt has been made to give a historical review of marine propulsion, and of the innumerable forms of screw proposed since 1836, only those are described which embody some intelligent idea.

The table of constants for disc-area and revolutions on p. 73 will, I think, be appreciated by those who have many screws to design. The complete series of model experiments upon which it is based is the work of Mr. R. E. Froude, and I am indebted to him for permission to make use of it.

There are many ways in which it is possible to tabulate experimental results, but after much consideration I believe that the table of constants which I have here given is the best that can be

devised, being independent of scale and equally accurate for all sizes of propellers. It is possible by means of it to design a screw which shall have maximum efficiency under any given conditions of indicated horsepower and speed; or if revolutions or diameter are so limited as to preclude the adoption of the most suitable dimensions, those may be selected which will be the best under the given conditions and the efficiency at once ascertained, provided only that the vessel is of such a form that the propulsive coefficient is not abnormal, and that the designer can correctly estimate the speed of the following current in which the screw works. More than this I do not think can be reasonably expected. No table can supply the place of judgment and experience.

SYDNEY W. BARNABY.

THE HOLLIES, CHISWICK MALL, W.
September 14th, 1891.

PREFACE

TO FIRST EDITION.

In preparing these lectures for the students of the Royal Naval College, I availed myself of information from various sources.

Now that they are published verbatim in book form, it becomes a duty as well as a pleasure to acknowledge the assistance thus received.

I am indebted first of all—and who that studies the subject of Marine Propulsion is not?—to Professor Rankine. Also to Mr. W. Froude, Mr. Bourne, Mr. White, Professor Osborne Reynolds, Mr. Sennett, Mr. Maginnis, and Mr. Seaton.

There is, however, much that is new.

The curves in Plate II., which enable the diameter, pitch, and speed of revolution of a screw suitable for any horsepower and any speed to be determined, are now made public for the first time.

For most of the new material, and especially for permission to publish these curves, and the method of producing them, I am indebted to Mr. Thornycroft.

It was my privilege to be associated with him in making some 550 experiments with model screws, and a considerable portion of these lectures is the result of knowledge thus obtained.

I feel some diffidence in putting forward in my own name information so acquired, as the credit of it is entirely due to Mr. Thornycroft, but after this explanation, any merit which may be found in the following pages will be attributed to the proper source.

SYDNEY W. BARNABY.

THE HOLLIES, CHISWICK MALL, W.
October 12th, 1885.

PREFACE

TO SECOND EDITION.

I HAVE purposely adhered to the original form of these lectures in preparing a second edition, although they are not so well arranged as they might have been, had they been written with a view to future publication.

With the exception of such textual corrections as were necessary, by reason of errors which had been overlooked in the first edition, they remain as they were delivered, and are largely supplemented by Notes in an Appendix.

I have to thank several correspondents who have pointed out some of these errors, and I shall be glad to be informed of any remaining uncorrected.

I am especially indebted to Mr. C. H. Wingfield, Assoc. M. Inst. C.E., for his careful revision of the new edition.

SYDNEY W. BARNABY.

THE HOLLIES, CHISWICK MALL, W.
January 10th, 1887.

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When the vessel is in motion at a regular speed the reaction $\frac{WS}{g}$ is equal to the resistance.

MARINE PROPELLERS.

CHAPTER I.

ERRATUM.

Page 19, line 16, *for* "P R" *read* "P R − V."

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MARINE PROPELLERS.

CHAPTER I.

FIRST PRINCIPLES.

THE principle upon which nearly all marine propellers work is the projection of a mass of water in a direction opposite to that of the required motion of the vessel.

If the weight of the mass of water acted upon by the propeller in pounds per second = W, and if the sternward velocity in feet per second imparted to it in relation to still water = S, then the reaction which constitutes the propelling force is $\frac{WS}{g}$ where $g = 32 \cdot 2$ feet per second; and this is independent of the form of propelling apparatus altogether. S is commonly known as the real slip, but will here be generally referred to as the rate of acceleration, or more shortly as the acceleration.

When the vessel is in motion at a regular speed the reaction $\frac{WS}{g}$ is equal to the resistance.

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So long as there is a resistance to be overcome by the propeller, there is no possibility of reducing the real slip or acceleration S to zero, except by making W, the weight of water acted upon, infinite.

When a propeller is to be designed for any given set of conditions, it is of the first importance that the relation between the mass of water acted upon and the acceleration imparted to it should be such that while the product $\frac{W S}{g}$ shall equal the estimated resistance of the ship, and the size and rate of motion of the propelling apparatus such as shall suit the conditions of the case, the economic result may yet be the best attainable, or may only fall short of the maximum by an amount which is calculable, and which it may be desirable to sacrifice in order to obtain other advantages.

The method of calculating the propulsive effect of the screw from the backward slip is that adopted by Rankine. Professor Greenhill has published some able papers on the theory of the screw,* in which he determines the propulsive effect as due to the reaction of the water to the rotatory motion given to it in the wake, and Professor FitzGerald has shown † that if the motion imparted to the water by a particular

* Trans. Inst. Naval Architects, xxix. p. 319.

† *Engineer*, August 22 and September 19, 1890.

screw is assumed to be a certain form of vortex, and a calculation be made of the power absorbed in producing such a vortex, it will be found to agree very well with the actual horsepower put through the screw. It is to be hoped that light will be thrown upon some remaining obscurities in the action of the screw by these independent investigations, but as they lead to the same conclusions as the older treatment, and as the latter is simpler and of more general application, it has been thought best to adhere to it.

Rankine has defined * the theoretical limit towards which the efficiency of propellers may be made to approximate by mechanical improvements, and has pointed out certain causes which make the actual efficiency fall short of that limit. He states that "if the propelling instrument be so constructed as to act upon each particle of water at first with a velocity equal to the velocity of feed"—that is the speed of the water entering the propeller—"and gradually increasing at a uniform rate up to the velocity of discharge, then the loss of work is the least possible." The oar, when a uniform force is applied to it by the oarsman throughout the stroke, approaches closely to this limit, as does also the screw-turbine propeller. (See p. 108.)

There is a certain quantity of work which

* *Engineer*, 1867, xxiii. p. 25; and 'Miscellaneous Scientific Papers,' edited by W. G. Miller, p. 544.

must be lost under all circumstances, and it is equal to the actual energy of the discharged water moving astern with a velocity S relative to still water. As this energy varies as the weight multiplied by the square of the velocity, if the quantity of water acted upon is doubled the loss from this cause is doubled, but if the acceleration is doubled the loss is increased fourfold. This explains why the hydraulic propeller, which is forced to act upon a much less area of column than the screw, appears at such a disadvantage when compared with it.

The causes of loss of work in propellers of different kinds may be thus summed up:—1st, Suddenness of change from velocity of feed to velocity of discharge. The radial paddle-wheel is defective in this respect. The feathering wheel, the screw, the Ruthven pump, and the oar are more or less exempt. 2nd, Transverse motion impressed on the water. Propellers which lose in efficiency from this cause are ordinary screws, which impart rotary motion; radial wheels, which give both downward and upward motion in entering and leaving the water; and oars, which impart outward and inward motion at the commencement and end of the stroke respectively. This loss is greatly reduced in the screw-turbine, and may be entirely avoided in the hydraulic propeller. 3rd, Waste of energy of the feed water. This is experienced by the hydraulic

propeller only as generally applied, and has been one of the causes of its inefficiency. It is not, however, a defect inherent to it, and has been avoided in some later applications.

CHAPTER II.

THE PADDLE-WHEEL.

As a propelling instrument the paddle-wheel is not inferior to the screw, but its speed of revolution is necessarily slow, and paddle engines are therefore larger, costlier, and heavier than screw engines of the same power. Until the introduction of the screw-turbine, it was the only propeller used for vessels of very shallow draught.

In order to ascertain the comparative value of the paddle and screw for towing purposes, H.M.S. *Rattler* and *Alecto*, the former a screw and the latter a paddle vessel of the same size and power, were lashed stern to stern. The *Rattler* towed the *Alecto* astern against the whole power of her engines at the rate of 2·8 knots per hour. Almost as much power can be developed with the screw when the vessel is towing as when she is running free, but this is not the case with the paddle-wheel. The engines of the *Alecto* could not get away, and were only able to develop 140 I.H.P., while the *Rattler* was developing 300.

It is difficult to frame rules for determining the proper area of floats for a given I.H.P. and speed of vessel, because so much depends upon

the position of the water surface in relation to the wheel when the vessel is at full speed.

There is usually a wave hollow in way of the wheel due to the motion of the vessel, and the action of the paddles is to cause the water to run towards them, and to produce a still greater depression of level in front of and below the wheel. Unless the vertical position is properly arranged the immersion of the floats will be insufficient at full speed, and the slip will be excessive. It is the practice of at least one eminent firm of shipbuilders to make model experiments in a tank, with the wheel in place, and revolved by clockwork at the proper speed, in order to ascertain the amount of the depression. The usual course followed in designing the wheels for a new vessel is to work from the nearest type, within the builder's experience, which has given good results.

Rankine gives a method of calculating the effective sectional area of a pair of feathering floats, which depends upon the general proposition already stated (p. 1), and which is common to all propellers.

If V = speed of vessel in feet per second;
*S = speed of centre of float relatively to the water in feet per second;
A = area of a pair of floats in square feet;
R = resistance of the vessel in pounds.

* $$S = \frac{100\,V}{100 - \%\text{ of slip}} - V.$$

Then, since $R = \frac{WS}{g}$ (see p. 1),

$$R = \frac{64 \times A(V + S)S}{32}.$$

$$\therefore A = \frac{R}{2(V + S)S}.$$

The formula is useful for purposes of comparison, as showing how the effective area varies with power, speed, and slip, and it should give a fair approximation to the area of float immersed at full speed; but as it is usual to make the top edge of the lowest float about awash when the vessel is at rest, the width will be increased above that theoretically necessary by an amount equal to the fall of water-level at the wheel.

In applying this formula to radial wheels, S should be taken as the speed of the lower edge of the float.

In radial wheels the number of floats should equal the number of feet in diameter, and the breadth of a float is usually from $\frac{3}{4}$ inch to one inch for each foot of diameter. In a feathering wheel the floats should be half as numerous and twice as broad as the floats of a radial wheel. The width of the wheel should be from one-third to one-half the breadth of the ship.

The diameter of the wheel is determined by the intended speed of the ship, the slip, and the number of revolutions considered most suitable,

generally from 20 to 30 per minute, but sometimes as many as 50.

Example :—

Speed of ship 15 knots	= say 1500 ft. per minute.
Slip 16 per cent.	= say 300 „ „
∴ Circumferential velocity of wheel	= 1800 „ „
Revolutions 30 per minute.	

$$\text{Diameter} = \frac{1800}{3 \cdot 14 \times 30} = 19 \text{ feet.}$$

The diameter would be taken at the centre of floats in a feathering wheel.

Fifteen to twenty per cent. is an average slip. The floats of a feathering wheel are constructed to cleave the water without shock. If *a*, Fig. 1,

Fig. 1.

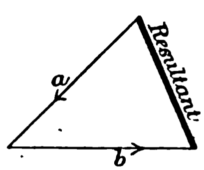
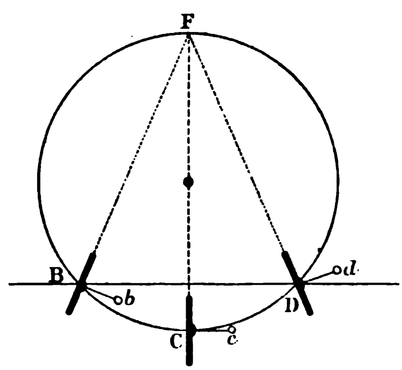


Fig. 2.



represents the direction of motion of a descending float, and the distance moved through by it in a given time if the vessel were stationary, and *b* equals the travel of the ship in the same time, then the actual path of the float is represented by the

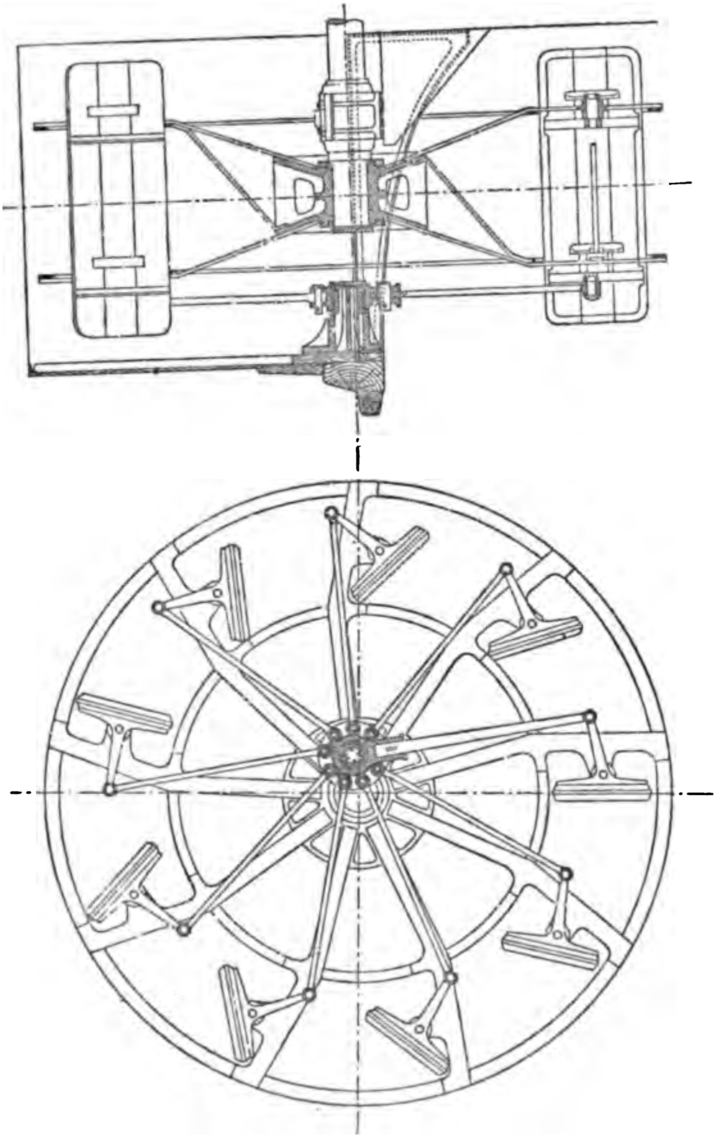


Fig. 3.

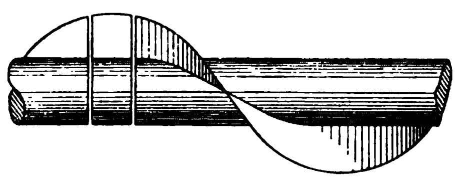
resultant. The plane of the float entering the water should coincide with this line. It is found that if a circle B F D, Fig. 2, be drawn through the centres of the floats, lines from the summit F to the points of intersection with the water-line B D will generally give the direction of the floats with sufficient accuracy. Levers, the lengths of which are about three-fifths of the depth of the float, are fixed perpendicularly at their centres B*b*, C*c*, D*d*. The centre of a circle, in the circumference of which the ends *b c d* of these three levers lie, is the centre of the excentric which produces the required motion of the floats. Fig. 3 represents the wheels of a vessel built by Messrs. Napier, Shanks, and Bell, and engined by Messrs. Rankin and Blackmore, of Greenock, to whom we are indebted for the illustration. The dimensions of the vessel are :—length, 260 feet; beam, 28 feet; draught, 5 feet 9 inches. The I.H.P. was 2680, and the speed 18¼ knots. The diameter of the wheel is 20 feet 6 inches over the floats, which are made of straight elm 9 feet 9 inches by 3 feet 6 inches. The revolutions were 47 per minute, and the slip 26½ per cent. The lowest float was immersed one inch from the top edge when at rest. The slip is rather high, and it is probable that a still better performance would have been obtained if the wheel had been rather more immersed.

CHAPTER III.

THE SCREW.

THREE strings wound spirally round a cylinder make a three-threaded screw. If instead of strings flat blades be wound edgewise, each having an edge soldered to the cylinder, then if a slice be cut off as shown in Fig. 4, there will be one piece of

FIG. 4.



blade attached to the slice if the screw have one thread, two pieces if two threads, and so on.

The "length of the blade" is equal to the length of the slice thus cut off, and the length of the cylinder necessary to contain one complete convolution of the blade is the "pitch" of the screw. The ratio of the length of the blade to the pitch is called the "fraction of pitch," a term more in use on the Continent than in this country.

It will be seen that the "pitch" of the screw has to do only with the form of the helix itself,

and has nothing to do with the velocity which may be imparted by it to the water. Theoretically, the blade is supposed to be of the form of a thin plate, as shown in Fig. 4, of equal thickness throughout and with both faces alike. The side of the blade which presses against the water when propelling the vessel ahead is called the "front" or "driving face"; the opposite side is the "back" of the blade. When speaking of the screw as a whole the nomenclature is commonly reversed, "in front of the propeller" standing for forward of the screw in relation to its position on the vessel, and "behind the propeller" meaning abaft it. These terms "in front" and "behind," which are apt to be confusing, will be avoided in this work.

FIG. 5.

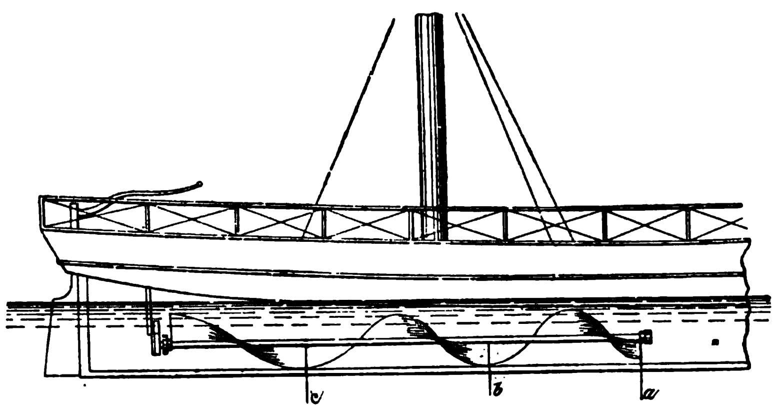


Fig. 5 shows a screw of "expanding or "increasing" pitch as originally proposed by Woodcroft. The length of cylinder necessary to contain

a complete convolution is not the same at all parts, and the pitch therefore is "variable." The first turn of the thread has a pitch equal to the length *a b*; the second has a longer pitch equal to *b c*. If a slice be cut off the cylinder (represented in Fig. 5 by the screw-shaft) midway between *a* and *b*, and of any length, it would be said to have a mean pitch equal to *a b*. A slice taken at a position midway between *b* and *c* would be said to have a mean pitch equal to *b c*, and similarly for any other part.

The edge of the blade which first meets the water is called the "forward" or "leading" edge, distinguishing it from the "after" or "trailing" edge.

The "disc area" is the area of the circle swept by the blade tips.*

The "developed area" or "blade surface" is the sum of the areas of all the blades, exclusive of the boss.

The "projected blade area" is the sum of the areas of the blades, exclusive of the boss, projected upon an athwartship plane.

It is convenient to use a term expressing the relation which the product of the pitch and revolutions of the screw per minute bears to the advance of the vessel through the water per minute. This term is "apparent slip."

* Rankine uses "effective disc area," which is the area exclusive of the area of the boss, but it is now usual to measure disc area as defined in the text.

If P = mean pitch;
R = revolutions per minute;
V = speed of ship in feet per minute;

then $\frac{P R - V}{P R} \times 100$ = percentage of apparent slip. In the great majority of cases $P R$ is greater than V, but it is occasionally equal to or less than V. The apparent slip is positive, nil, and negative in the three cases respectively. The real slip or acceleration of the water S (see p. 1) is not measurable from the pitch and revolutions of the screw. A propeller placed at the stern of a vessel is in a current having a forward mean velocity U relative to still water caused by the friction of the skin of the vessel. While the vessel advances at a velocity V through the water, the screw advances at a less velocity $= V - U$. If the apparent slip is nil then the real slip of the screw is equal to U. Conceive a propeller of increasing pitch which shall be so designed that the pitch of the forward edge multiplied by the revolutions per minute shall be equal to the speed of the propeller through the water, that is $V - U$, and the pitch of the after edge multiplied by the revolutions equal to V, then the mean pitch, as already explained, would be called $\frac{V + (V - U)}{2}$. The late Mr. Froude showed that under these circumstances apparent negative slip is possible. He describes an ideal case in which the whole of the resistance

of a vessel consists in skin friction, wave-making and other factors being excluded. The dynamic equivalent of the propulsive force employed in keeping her in motion is found in the frictional wake, and a propeller which should pervadingly operate upon the wake in such a manner as to bring it gradually to rest, would, in thus neutralising it, maintain the propulsive force, and, given established motion, a theoretically perfect propeller, quite clear of the ship's stern, would maintain that motion and exhibit apparent negative slip equal to half the forward mean velocity of the wake at the point where the propeller operated.

In this case it is clear that apparent negative slip results from the fact that while a sternward velocity is supposed to be imparted to the water equal to the product of the number of revolutions multiplied by the pitch of the after edge of the screw, the mean pitch of the screw itself is nominally less than the pitch of the after edge. Negative slip would disappear in this particular case if the speed of advance of the screw were calculated from the after pitch instead of from the mean.

Apparent negative slip is sometimes exhibited by screws of uniform pitch when the ratio of pitch to diameter is small. The twin screws of H.M.S. *Collingwood*, with a pitch-ratio of 1·5, gave 1·26 per cent. apparent negative slip, and this was increased to 2·56 per cent. when the pitch-ratio was reduced to 1.

Numerous explanations have been given of the phenomenon of apparent negative slip, but none of them can be accepted as satisfactorily accounting for its occurrence in the case of well-formed uniform-pitch screws, the pitch of which has been carefully verified as in the case of the *Collingwood.*

It has been suggested that the blades twist or spring under the pressure of the water, which would have the effect of increasing the pitch, and that they recover their shape when the pressure is relieved, so that measurements taken after the trial even would be misleading.

If this were the case screws with thin blades would be most likely to show negative slip, but it is, on the contrary, generally met with in screws with very thick blades where springing would be least likely to occur. Moreover, it might be expected that the metal would soon give way by fatigue if it were distorted by the pressure, as the fluctuations are very great and very rapid.

What has been said about the effect of thick blades lends force to another suggestion, which is that it may be attributed to the effect of the round back of the blades. It has been explained, p. 13, that pitch is measured on the assumption that both sides of a blade are alike. The effect of the round back of the blade must be to increase the effective pitch and to tend to reduce the apparent slip, although it is difficult to say by how much, and it

is not, in the author's opinion, sufficient to account for the *Collingwood* results.

It has been pointed out that the intermittent action of the blades passing through the dead water abaft the stern-post of a full formed ship reduces apparent slip and may even cause it to change sign, but this does not affect the case of the ship referred to, which is a fine ship, and the screws being twin, are not behind a stern-post.

Another suggestion has been that the motion of the particles in the wave which followed and enveloped the stern of the ship might produce apparent negative slip. All attempted explanations based upon the effect of dead water and following current alone, apply only to a screw of increasing pitch, and this has already been dealt with (p. 16). They are quite inapplicable to a screw of uniform pitch, and the same thing is true of the theory based upon the circular motion of the particles of water in waves. The waves referred to are presumably those made by the vessel, because it cannot be contended that negative slip results from the screw working among free waves. It is well known that the contrary effect is produced, the slip is increased, and the most favourable condition for the exhibition of apparent negative slip is still water. If a wave follows the ship, and its energy can be made use of in any way by the screw, that wave has previously been created by the ship from which the energy has been robbed, and can be only

partially restored. Could it all be given back, and could the whole of the energy of the frictional wake be utilised by the screw without loss, there would still be no surplus thrust to keep the vessel in motion. It is certain that there must be a stream of water left behind by the screw having sternward motion relative to still water. How is it then that notwithstanding this necessity, apparent negative slip is occasionally obtained with screws of uniform pitch? In a paper read before the Institution of Civil Engineers on the screw propeller,* the author gave what seems to him a satisfactory explanation based upon a proposition recently enunciated by Mr. R. E. Froude,† to the effect that the slip or acceleration S of the water in the race was always in excess of the slip of the screw (P R.–V) Mr. Froude showed that if no rotation were produced in the race by the propeller, a limiting case was reached in which one-half of the whole acceleration would be produced forward of the propeller and one-half aft of it. The water forward of the propeller was affected before it was actually in contact with it, and would run towards the screw, meeting it with a velocity, as regards still water, of $\frac{S}{2}$, and the action of the propeller upon the water while in

* 'Proceedings of the Institution of Civil Engineers,' cii. p. 74.

† 'Transactions of the Institution of Naval Architects,' 1889, xxx. p. 390.

contact was to accumulate pressure which had the effect of increasing the acceleration of the race after it had left the propeller. This is perhaps more easily understood if we suppose the propeller to be stationary in an ocean moving with a velocity V. At some distance forward of the propeller the water will be advancing to meet it at a velocity V. On nearing the propeller, the water is accelerated by its sucking action, and meets it at a velocity $V + \frac{S}{2}$. The length of the blades may be supposed to be so small that no appreciable change in velocity can take place in the stream while actually passing through them, but after leaving them the speed of the stream is further accelerated up to the final speed $V + S$. The mean speed of stream in which such a propeller works is therefore $V + \frac{S}{2}$, and the real slip of the propelling apparatus, which is a measure of its efficiency, would be $\frac{S}{2}$, but the speed of the race is $V + S$, and the real slip of the water is S. Such a propeller might show a considerable amount of apparent negative slip if placed behind a ship and in a following current, a condition which is of course essential, as if it were propelling a phantom ship (see p. 58) the apparent slip would be the same as the real slip, viz. $\frac{S}{2}$.

It is impossible to make an open screw which

shall not rotate the water, but the finer the pitch is the less the rotation will be.

Mr. Thornycroft has shown that the relation between the amount by which the race is accelerated forward and aft of the screw respectively may be expected to depend upon the amount of the rotation produced. A screw of coarse pitch-ratio which will rotate the race considerably, will produce a large proportion of the whole acceleration before the water reaches the screw, leaving only a small part to be imparted abaft it. When the rotation is a maximum, the whole acceleration is produced by suction, and the speed of the stream on meeting the propeller is $V + S$, in which case the real slip of the screw is equal to the real slip of the water S. All open propellers, by which is meant propellers which are not confined in a casing like the screw-turbine, occupy some intermediate position, and are working in a stream with a velocity varying between $V + S$ and $V + \frac{S}{2}$, depending upon the greater or less rotation of the race. Hence the finer the pitch-ratio the more favourable would be the conditions for obtaining apparent negative slip, and it is found to be invariably the case that it only occurs under these conditions, and that it may be increased by still further reducing the pitch of screws exhibiting it (see p. 16).

There are a number of pitchometers made which will measure the pitch of a screw with sufficient

accuracy if it is uniform. They are not to be depended upon for the measurement of increasing pitches. The operation may be performed without instruments, as follows :—

Strike an arc of a circle *a b*, Fig. 6, concentric with the axis of the propeller, upon one of the blades with any radius R. Divide the arc into a number of equal intervals, as 1, 2, 3, 4, 5. Measure off the same number of intervals upon a base-line *x y*, Fig. 7, making them equal to the developed length of the intervals upon the arc. Measure the ordinates at each interval on the arc from a plane perpendicular to the axis of the propeller, and lay off the ordinates 1 1′, 2 2′, 3 3′, &c., at the corre-

Fig. 6.

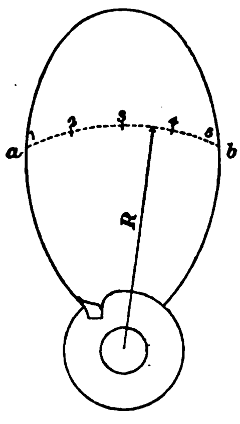
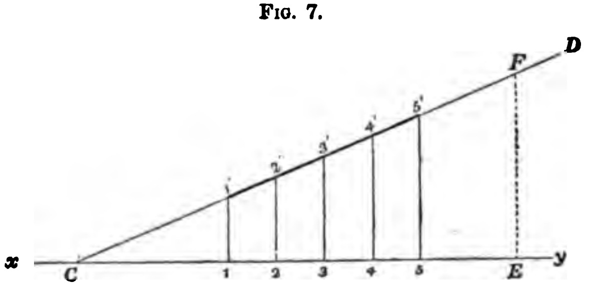


Fig. 7.



sponding intervals on the base-line. Draw a line C D through the points 1′, 2′, 3′, &c., and produce

it to cut the base in C. If the pitch is uniform C D will be a straight line. Measure off from C a length C E equal to the circumference of the circle of R radius, and erect a perpendicular at E cutting C D at F. E F is the pitch of the screw. If the pitch is not uniform the points 1′, 2′, 3′ will lie in a curve. Tangents must be drawn to the curve at the extremities 1′ and 5′, and the pitch of each measured separately, see Figs. 8 and 9, and the

FIG. 8.

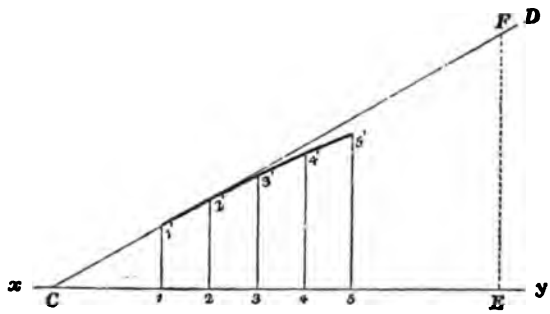
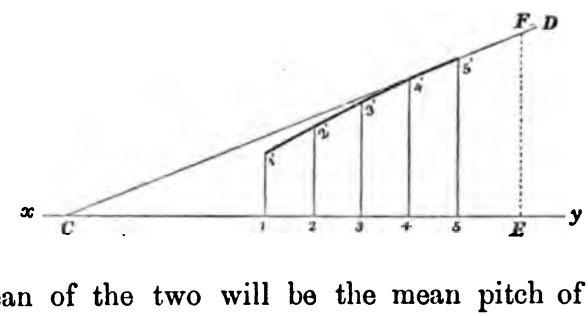


FIG. 9.



mean of the two will be the mean pitch of the screw. Measurements should be made of each blade and at a number of different radii, and the

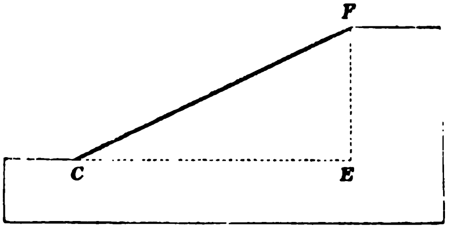
mean of all the readings taken as the mean pitch. If the pitch is uniform, and great accuracy is not required, choose any two points on the arc *a b*, Fig. 6, such that radial lines from them to the axis subtend an angle of 30°. The difference between the length of the ordinates of these points to a plane at right-angles to the axis, measured in inches, is equal to the pitch in feet.

The pitch of small model screws used for experimental purposes can be most readily obtained as follows:—Make a cylinder of wood of a diameter about two-thirds that of the propeller. Fit a short mandril to represent a piece of the shaft into the boss, and pass the mandril through a hole in the axis of the cylinder which has been bored to fit it. Wrap a sheet of paper round the cylinder, securing it with an elastic band, and cut the edge accurately to fit the face of the blade. The direction of the axis F E should be marked upon it. When the paper is taken off the cylinder and unrolled, the edge which fitted the face of the blade will form a straight line as C F, Fig. 10, if the pitch is uniform, and if D be the diameter of the cylinder, then $\frac{\text{F E}}{\text{C E}} = \frac{\text{Pitch}}{\text{D} \times 3 \cdot 14}$. If the blade is not of uniform pitch it will form a curved line, and the pitch of the leading and after edge must be obtained by drawing tangents to the extremities of the curve as already described.

It is sometimes useful to be able to estimate

roughly the pitch of a screw at sight. This may be done by observing at what radius the blade makes an angle of 45° with the axis; the pitch is equal to the circumference at this radius.

FIG. 10.

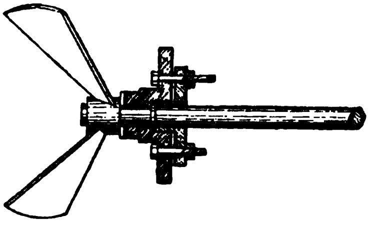


The screw was brought into successful operation as a propeller by Ericsson and Smith in 1836. The *Archimedes*, a screw vessel of 237 tons burden, was built by the latter in 1839. The screw used was a single threaded helix of one complete convolution. A double thread of half a convolution was afterwards tried and found to be an improvement; but the best result was obtained with two threads and one-sixth of a convolution. The Earl of Dundonald in 1843 patented a propeller with the blades thrown back as shown in Fig. 11, the object being to counteract centrifugal motion of the water, supposed to be caused by the rotation imparted to it by the screw.

When a propeller is not sufficiently immersed to prevent it from drawing down air, it is probable that centrifugal action takes place, and that the column of water takes the form of a cone with the

screw for an apex; but when no air gets to the propeller, observation of its action in the phosphorescent water of tropical seas appears to show that an ordinary screw does not disperse the water, but

FIG. 11.



projects a column more or less cylindrical, and having the appearance of a twisted rope, the strands of which unravel themselves as they pass astern.

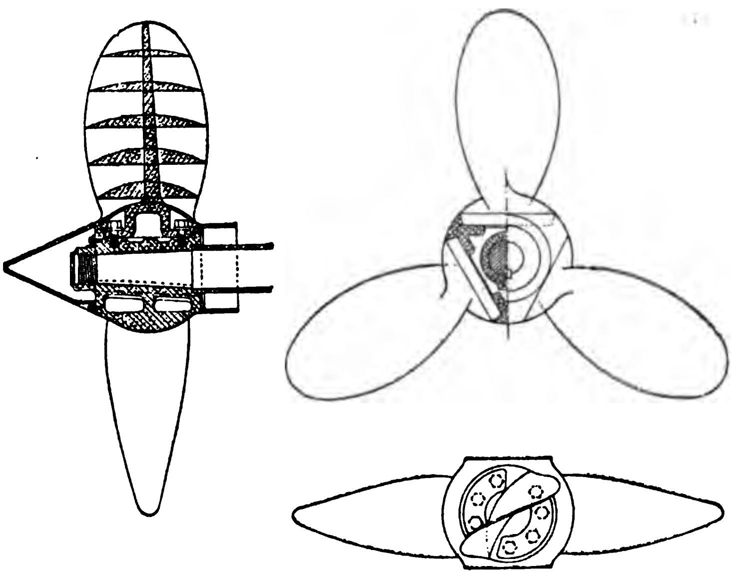
In 1849 Robert Griffiths patented a self-governing propeller, which he thus describes:—"If the screw moves with greater velocity than usual, the increased resistance of the leading edge shall correspondingly increase the pitch, thus increasing the resistance and bringing down the revolutions."

The propeller chiefly associated with his name is shown in Fig. 12, which represents the form now adopted in the British Navy. The principal feature in the Griffiths screw is the large boss, which, while not impairing the efficiency, enables the blade to be fixed in such a manner that the

* See a paper by M. Marchal in the 'Transactions of the Institute of Naval Architects,' 1886, xxvii. p. 238.

pitch can be readily altered. This is an important consideration, as it is difficult to fix upon exactly the right pitch in designing a propeller to run at a given number of revolutions.

Fig. 12.

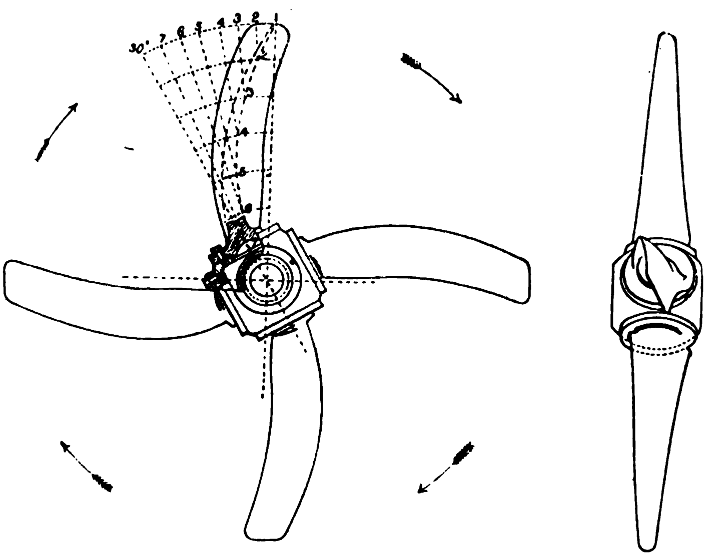


The Hirsch screw is shown in Fig. 13. It has an increasing pitch, and the propelling surfaces are so formed as to throw the water somewhat towards the axis.

The Mangin propeller, Fig. 14, consists of two narrow-bladed screws set behind one another on the screw-shaft with a space between them. It is supposed not to rotate the water so much as other screws.

Rigg's propeller had a fixed screw or guide-blades placed behind the revolving screw, with the blades set at the reverse angle, so as to take the

FIG. 13

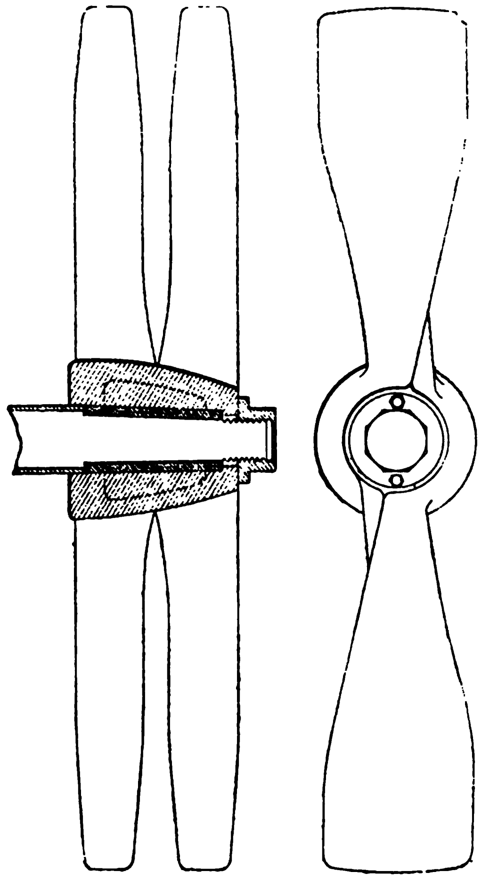


rotation out of the water and leave it moving directly astern. Rankine and Napier patented a modification of this idea in the form of a twisted rudder, of which the part above the screw-shaft bends in one direction and the part below in the opposite.

Screws have been tried with the pitch in the centre less than the pitch at the circumference, so as to allow the central part simply to follow up the water.

The Thornycroft screw, Fig. 15, which has proved very successful, has an increasing pitch at the middle of the blade, but it gradually becomes

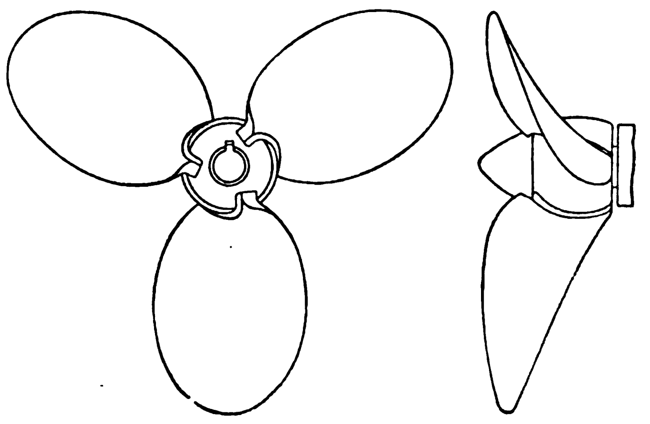
Fig. 14.



uniform towards the root and towards the tip, the reason being that at the root the rotation is already excessive, and it is consequently not advisable to

increase it by increasing the pitch, and towards the tip, if it is attempted to accelerate the water too much, it escapes round to the back of the blade.

FIG. 15.

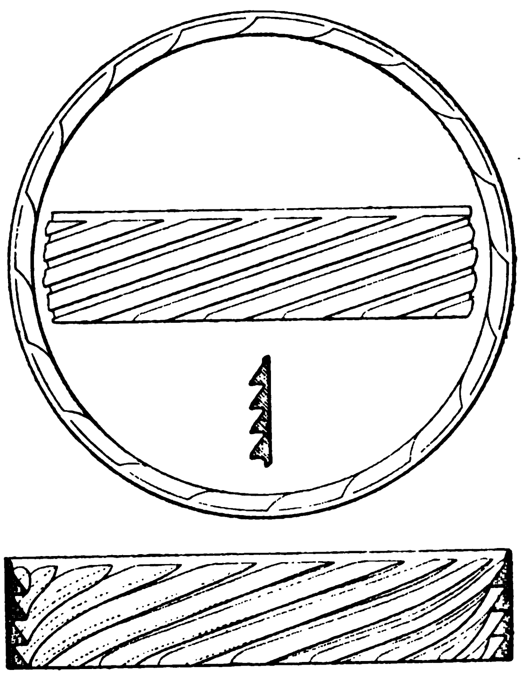


The blades are also thrown back after the fashion of the Dundonald propeller, but, instead of being straight, they are convex on the driving face.

An ingenious attempt to make use of a certain amount of energy said to be wasted by the screw has been made recently by MM. des Goffes and de George. The inventors say that the viscosity or resistance to rupture of the water in which any helical propeller revolves, causes an appreciable current to be set in rotation just beyond the tips of the screw-blades. It is contended that a series of helical surfaces, opposed in direction to those of the screw proper, will receive a thrust from the revolving ring of water, which can be utilised for pro-

pulsion. The "Antispire," as it is called, can be placed around a propeller of any form. It is shown in Fig. 16. The only experiments with the apparatus which the author has witnessed were made

FIG. 16.



in a tank, and as there was no motion through the water it was not possible to tell how much the result would be affected by the friction of the ring itself, but a large increase of thrust was found to be produced by it, and it is possible that the friction of the surfaces would be more than counterbalanced and a real propulsive force exerted. It is reasonable to suppose that there is an unutilised reservoir

of work in revolving currents external to the propeller disc. Rings or bands without helical blades have been used for protecting screws and for giving increased manœuvring power, but these plain rings do not increase the thrust or the efficiency of the screw, and the addition of the blades would certainly be found advantageous in such cases, and may be worthy of a still wider application.

In some double-ended ferry-boats, both in this country and America, screws have been placed at both ends of the vessel, for what appear to be sufficient reasons connected with the service which they have to perform. In the well-known Mersey boats there are four screws, but in a new ferry-boat, the *Bergen*, built in America, two only are employed, one forward and one aft, driven by the same shaft—an arrangement which appears to be inferior. The forward screw of the *Bergen* is estimated to augment the resistance of the hull by 23·5 per cent., and its propelling efficiency is only 43 per cent. of that of the after screw.

There is no doubt that the best position for a screw is at the stern. As a vessel moves through the water the friction of the sides and bottom imparts motion to a layer of water which increases in thickness towards the stern, so that a considerable quantity of water is left with a motion in the same direction as the vessel. If the screw works in this water it is able to recover some of the energy which has been expended by the ship in

giving it motion. The speed of this wake, which Rankine estimates may be as much as one-tenth of the speed of the vessel, depends not only upon the form, but upon the nature and extent of the surface. It would not be desirable to increase the volume or the velocity of a wake for the purpose of improving the efficiency of the propeller, because this very surface friction proves to be the largest portion of the resistance of a ship at moderate speed; but as it is a necessity that there should be a wake, it is a distinct advantage to place the propeller in it and allow it to utilise as much as possible of the energy it finds there.

This frictional wake must not be confused with dead water, which is water eddying behind a bluff stern, and which has acquired the full velocity of the vessel. When once the speed of the vessel has been imparted to this water, not much energy is wasted in maintaining it. If it is drawn out by the screw, fresh water must take its place, and there will be a continual drain of energy from the ship, as the inflowing water must in its turn have the full forward velocity imparted to it. Dead water is almost a thing of the past, and is met with only in the case of very full ships.

If a screw is placed behind a stern so bluff that the supply of water is impeded, it will draw in water at the centre of the driving face, and throw it off round the tips of the blades like a centrifugal

pump. The effect upon the ship is then peculiar. Sir Frederick Bramwell has described a vessel which went astern whichever way the screw was driven, the reason being that a very bluff stern caused the screw to act as described, and a loss of pressure was produced upon the stern of the vessel.

It is very important that a propeller should have sufficient immersion, since if it breaks the surface of the water the efficiency is reduced to a remarkable extent (see Plate 1); but if it is sufficiently far below the surface to prevent it from drawing air, any further immersion within the limits that can practically be obtained is of little value. The speed with which water can follow up the blades of a screw depends upon the head of water over them, but when air is excluded the equivalent of a head of 30 feet is supplied by the atmosphere, and this being elastic and having practically no inertia to resist sudden motion, its pressure is more effective than that of a column of water of equivalent weight.

It is probable that the inequality in the onward motion of the layers of water forming the frictional wake, accounts to some extent for the vibration caused by the screw, since each blade in revolving meets with an alternately diminished and increased resistance. An ingenious mechanical contrivance was invented by Griffiths, by which the blades were made to adjust their pitch to suit the resist-

ance, the pitch of a blade being reduced when passing through the upper part of the circle, and increased when passing through the lower part. The apparatus would not stand much wear and tear. Unless a propeller has a good running balance it will tend to cause vibration. To insure steadiness when revolving at high speed, it is necessary that each blade should be of the same weight, and that the centre of gravity of each should be at the same distance from the axis of the shaft.

There is a disadvantage connected with an inclined screw-shaft, which has been generally overlooked. The result of depressing the end of a shaft is to cause the effective pitch to vary through every part of the revolution. If the inclination be supposed to be 45° for example, that part of the blade which is intended to have a pitch of three diameters, has, in reality, an effective pitch varying from nothing to infinity. It is, of course, obvious that the pitch of the blades in relation to the axis is unchanged by any alteration in the direction of the shaft, but whatever the pitch in relation to the axis may be, if the axis were to pass vertically out through the bottom of the ship, the virtual or effective pitch, measured in the direction of motion, is nil. If a screw does not move along but has a motion of rotation only, the resistance of the water to the blades is the same whatever be the direction of the shaft, but if the propeller be

allowed to move along while at the same time it be constrained to move horizontally, the shaft being inclined to the horizontal, then the resistance of the water to the blades is not uniform, but varies over every part of the revolution. This will perhaps be made clearer by an examination of the phases through which a blade passes during one revolution. It is convenient and suitable to consider the action of a screw as similar to that of an inclined plane moving past the stern. In Fig. 17 the full line represents the upper blade as a plane moving from port to starboard, the dotted line represents the lower blade as a plane moving from starboard to port. In Fig. 18 the shaft is horizontal, and the full line shows the blade going down, and the dotted line the blade coming up. In Fig. 19 the shaft is inclined at 45°, the full line again shows the blade going down, and the dotted line the blade coming up. Now as the ship moves forward the water flows to the screw in approximately horizontal lines, and the blade which at one part

FIG. 17.

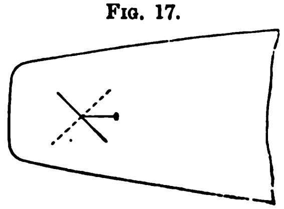


FIG. 18.

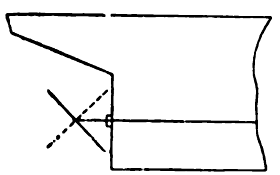
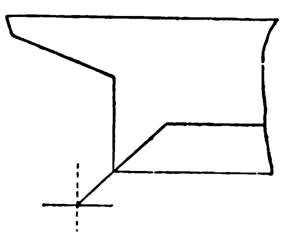


FIG. 19.



of the revolution is edgewise to the water, at another is square on to it, and the result is a succession of shocks causing vibration. Another way of looking at it is this:—A particle of water meeting the ascending blade has its motion relative to the vessel arrested completely, while a particle first meeting the forward edge of the descending blade would require to have its velocity infinitely accelerated in a horizontal direction to enable it to escape from under the blade. This is what is meant by saying that in the above example the effective pitch varies from nothing to infinity during each revolution.

The racing of screws is due to either of two causes. If the propeller breaks the surface of the water as the stern rises in a seaway, it will draw air down, and the resistance is immediately very much reduced. Referring to Plate 1, where the thrust is shown at different revolutions of a propeller, both when completely immersed and also when splashing, it will be seen that in the former condition a thrust of 11 lbs. is exerted at 680 revolutions. When air is drawn down, the same thrust is exerted at 1000 revolutions, so that this propeller, if delivering a constant thrust, would vary its revolutions very rapidly from 680 to 1000 if alternately raised and lowered as in the action of pitching. But it is not only when the screw breaks the surface that it will race. If a vessel is among waves, racing may occur, although the screw may

not be emerged at all. Mr. Froude pointed out that this was probably due to the circular motion of the particles of water in waves. There is no real motion of translation in waves, the water which is travelling in one direction at the crest returns in the opposite direction in the trough. This circular motion extends to some distance below the surface, and a screw finds the resistance of the water augmented or reduced, as it is beneath the trough or crest of a wave, and reduces or increases its speed accordingly.

A screw causes lateral motion of the stern of a vessel which has to be counteracted by the rudder. This effect is very much greater when going astern than when going ahead, but the cause is not the same in the two cases. When going ahead Professor Osborne Reynolds has pointed out that the onward motion of the frictional wake is very different at the surface and at the keel. He agrees with Rankine that the mean speed of the wake in the case of a vessel of fairly fine form may be 10 per cent. of the vessel's speed, but thinks it varies from 20 per cent. at the surface to nil at the keel; the upper blade of the screw therefore experiences more resistance than the lower, and tends to drive the stern round. If the screw is right-handed, and does not draw air down, it will tend to cause the vessel to carry a starboard helm in order to maintain a course. If there is air in the wake, caused for example by the vessel being

at a light draught of water, the effect is reversed, the lower blade predominates, and port helm must be carried. The natural effect of the screw may also be neutralised or even reversed if there is a broad counter over it and a large rudder, especially if, as is often the case, the part of the rudder behind the upper blade of the screw is larger in area than the part behind the lower blade. The reaction of the stream of water thrown from the upper blade upon the counter and upper portion of the rudder is greater than that of the stream thrown from the lower blade in the contrary direction upon the lower portion of the rudder, and may necessitate the carrying of a port helm with a right-handed propeller. Professor Reynolds states that a right-handed screw without air always bears considerably on the port side of the stern-post, even when the ship carries a port helm, showing that the effect on the hull and rudder more than counterbalances the effect on the screw. In the *Engineer* of October 1st, 1886, was published a diagram showing the port and starboard strains upon a rudder as recorded automatically by Maginnis' "Rudder-graph." The screw was right-handed, and the diagram showed clearly that the ship required a starboard helm to keep a straight course.

When the screw is reversed, and the vessel has gathered stern way, the propeller has a much greater influence upon the course of the ship than

when going ahead. In the latter case the influence is always very small; in the former it is often great. The engines will be observed to have a great tendency to race when going astern. The screw is then drawing air, and the upper blades suffer most, so that the lower blades experience the most resistance, and drive the stern round. A right-handed screw tends to move the stern to port, and a left-handed one to starboard.

When a screw is suddenly reversed, and before the headway is off the vessel, the action of the rudder is not to be depended upon. The following is an extract from the report of the Committee appointed by the British Association to investigate the effect of propellers on the steering of vessels.

British Association Report, 1878.

"It is found an invariable rule that during the interval in which a ship is stopping herself by the reversal of her screw, the rudder produces none of its usual effect to turn the ship, but that under these circumstances the effect of the rudder, such as it is, is to turn the ship in an opposite direction from that in which she would turn if the screw were going ahead. The magnitude of this effect is always feeble, and is different for different ships, and even for the same ship under different conditions of loading. It also appears that, owing to the feeble influence of the rudder over the ship during the interval in which she is stopping, she is then at the mercy of any other influences that may act upon her. Thus, the wind, which always exerts an influence to turn the stem of the ship into the wind, but which influence is usually well under control of the rudder, may, when the screw is reversed, become paramount, and cause the ship to turn in a direction the very opposite of that which is desired.

"Also the reversed screw will exercise an influence, which

increases as the ship's way is diminished, to turn the ship to starboard or port, according as it is right or left-handed, this being particularly the case when the ship is in light draught. These several influences—the reversed effect of the rudder, the effect of the wind, and the action of the screw—will determine the course the ship takes during the interval of stopping.

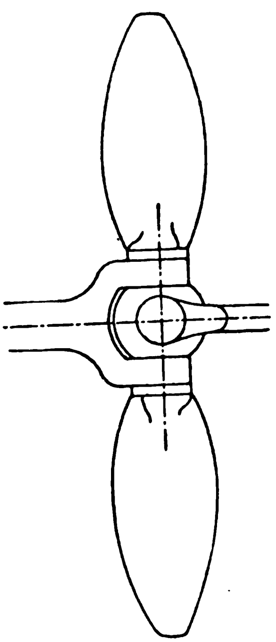
"They may balance, in which case the ship will go straight on, or any one of the three may predominate."

Notwithstanding that a screw has a tendency, as just described, to produce sideways motion of the stern, and so to cause a vessel to deviate from a straight course, it yet offers considerable resistance to lateral movement produced by external causes. The pressure on the blade which is moving in the same direction as that in which the stern of the vessel is turning is increased, while on the blade moving in the opposite direction the pressure is reduced, that is to say, if the screw is right-handed and the vessel is under port helm, the stern, consequently, travelling to port, the resistance of the lower blade which is moving towards the port side will be increased, and the resistance of the upper blade, which will be moving towards the starboard side, will be diminished, because the one is meeting the water, and the other is receding from it. The change of pressure will be proportional to the square of the angular velocity of the stern. The irregular pressure causes the vibration frequently noticed when a screw vessel is rapidly turning. This resistance to lateral motion is not without value, because if it is removed the condition of a

vessel moving in a straight line is one of instability. If the vessel makes the least angle to the direction in which she is moving, the excess of pressure due to undisturbed water at the bow tends to increase the divergence, and this tendency is resisted by the propeller. It seems probable that a vessel never maintains a line of advance in the exact direction of its axis, but always at a small angle with it.

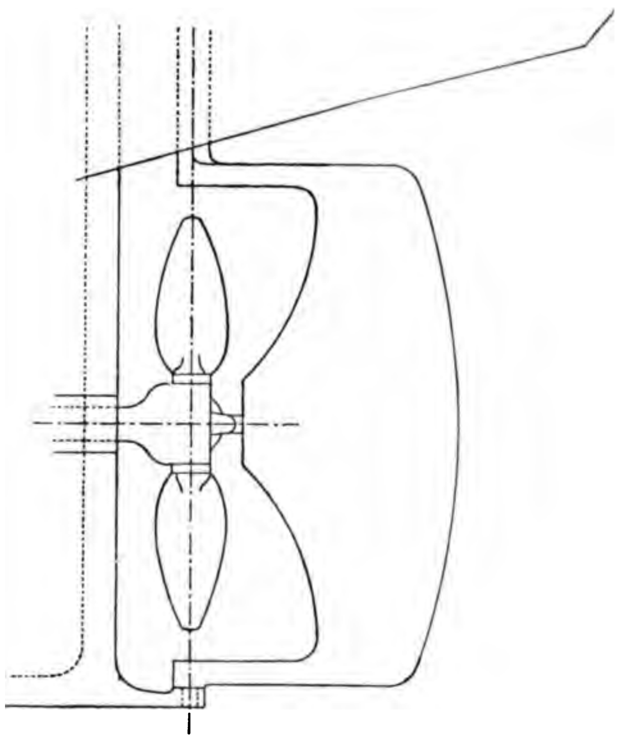
A very ingenious propeller has lately been patented by Mr. F. H. White, which, while retaining the advantage of rigid blades in resisting lateral motion when a vessel is on a straight course, is so arranged that the blades are under control and can be feathered in such a way as to cause them not only to offer no resistance to turning, but to actively assist in it. It is in fact a steering propeller of a very simple description. It is illustrated in Figs. 20, 21, 22. The screw-shaft terminates in a universal joint which connects it with an extension of itself in the form of a smaller shaft or tail-piece. The joint can be made in several

FIG. 20.



forms, but the simplest is shown in Fig. 20 where the centre or connecting piece has four arms at right angles to each other. The blades are rigidly fixed to the two arms which are di-

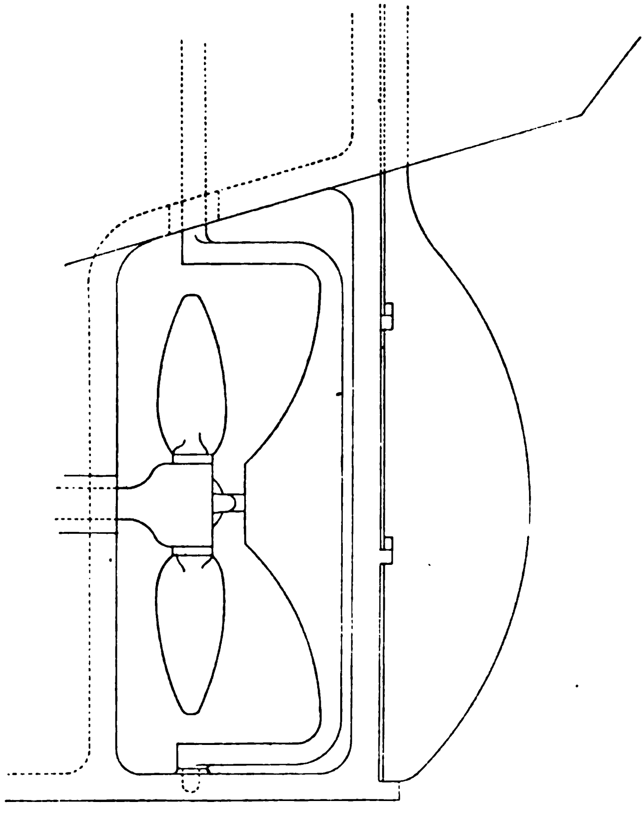
FIG. 21.



rectly held by the jaws of the main shaft. See Fig. 20, which shows the screw in elevation. Any movement of the tail-piece causes an alteration in the pitch of the blades, an increase in the pitch of one blade being accompanied by an equal

decrease in the pitch of the other. The effect of the change of pitch is that the stern of the vessel is forced either in one direction or the other,

Fig. 22.



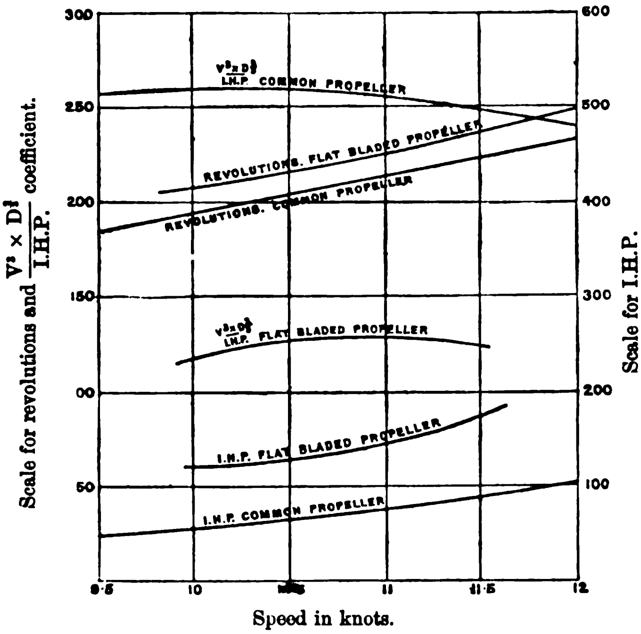
according to which side the tail-piece is moved. When it is desired to change the course of the vessel to the right, the tail-piece is moved to the right, its manipulation being thus similar to that

of the rudder to which it may be attached as shown in Figs. 21 or 22. The joint is covered with a light casing as shown in Fig. 21. The screw may have two, three, or four blades. A characteristic which adds much to the practical value of the design, is that the feathering of the blades is greatly assisted by the action of the water-flow itself, as any alteration in the course of the vessel tends to change the pitch of the blades in such a manner as to bring the tail-piece into that position which would of itself cause such an alteration, so that after having initiated the feathering motion, it may be anticipated that the tail-piece will have little more to do than to control and regulate it.

When auxiliary steam power has to be applied to sailing vessels, it is best, if possible, to arrange the screw so that it can be lifted out of the water. If disconnected and allowed to revolve it causes considerable resistance. If it cannot conveniently be lifted, it is advisable to use a screw with two narrow blades, which can be set up and down in a line with the stern-post, and feathered fore and aft by internal mechanism as arranged by Bevis. With a view of reducing the drag under sail to a minimum, Messrs. Thornycroft tried a propeller with flat blades, which could be feathered in line with the shaft for sailing, and set at an angle with it for steaming, but it was found to be a very inefficient propeller, requiring just double the

horse-power for a given speed as was needed for an ordinary screw. Fig. 23 shows the results obtained with it, compared with those given by an ordinary screw by which it was afterwards replaced.

FIG. 23.



Comparative trial with common and with flat-bladed propellers.

Vessels intended for towing require large screws, because, if the screws are designed to work at their best efficiency against the small resistance of the tug alone, the slip when towing will be excessive, and will cause an undue waste of power. It is desirable to so design the propeller that it shall give a maximum return at the speed which the

tug may be expected to attain when towing an average load.

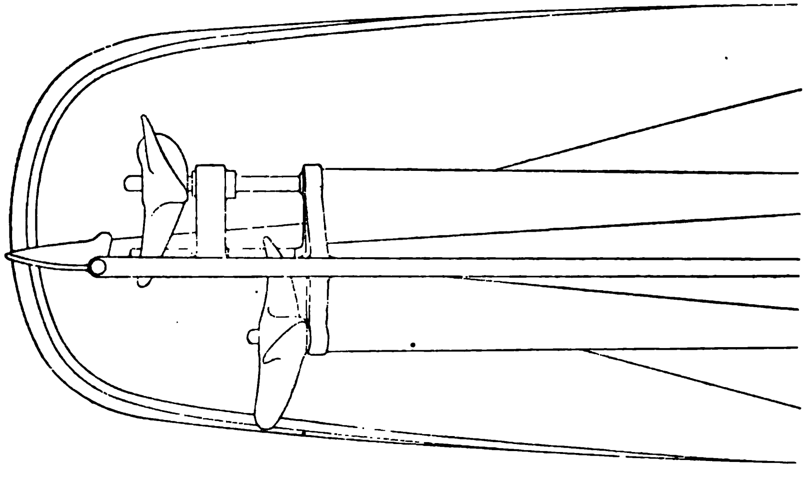
Screws for electric launches labour under the disadvantage of having to run at an exceptionally high speed of revolution. The blades should be very thin and sharp, and two blades will give less resistance to turning with a given surface than three. It is difficult to estimate the pitch necessary to give a required number of revolutions, because it is a peculiarity of the motor, that the slower the rate at which the screw turns, the faster the power is run down, and *vice versâ*. It may be compared in this respect to the steam siren which uses a large amount of steam when revolving slowly, and the more rapid the rate of turning the less is the quantity of steam passed.

The fastest running screws of which the author has had experience were made by Messrs. Thornycroft, for the Howell torpedo. They are 6 inches in diameter, and run at 5000 revolutions per minute, driving the torpedo at 30 knots.

Twin screws possess very many advantages over a single screw, and are quite as efficient. In very fine ships the length of the outside shafting becomes a serious consideration, and the necessary supports add to the resistance. In order to reduce this inconvenience to a minimum, the well-known firm of Rankin and Blackmore introduced an arrangement of twin screws with overlapping discs in the tug *Otter*, built in 1876. One propeller

was set in front of the other, the blade-tips passing through an aperture in the dead wood. Figs. 24 and 25 show twin screws as fitted in the *Buzzard*, a small coasting steamer belonging to Mr. John Burns. The arrangement has proved successful,

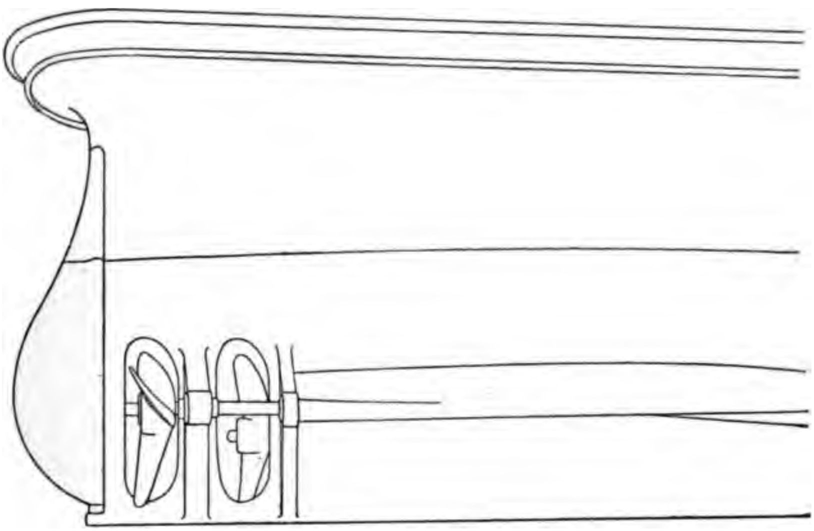
FIG. 24.



and has been frequently adopted, the most notable examples being the s.s. *Teutonic* and *Majestic*, built by Messrs. Harland and Wolff. The screws of the *Teutonic* are 19 feet 6 inches in diameter, and the distance between the shafts is 16 feet. One screw is 6 feet 3 inches behind the other. They are right and left-handed, and turn outwards. In a twin screw torpedo boat built by M. Normand, with overlapping screws, both are arranged to turn the same way. The overlapping blades thus cross one another, and the water thrown up by the ascending

blade of one screw is met by the descending blade of the other, and the slip is reduced. It is found that when so arranged, the aftermost propeller turns slower than the foremost one, the contrary being of course the case when they turn in opposite directions.

FIG. 25.



Triple screws have not as yet been widely introduced. An interesting series of comparative trials with twin and triple screws was made by M. Marchal at Lorient, and described by him in a paper read at the Institution of Naval Architects in 1886. His summing up of the results is that "three screws are, from the point of view of speed, very nearly equivalent to two screws of the same propulsive surface and immersed to the same

depth, when the most favourable position is chosen for each system." * Recent examples of the application of three screws are the French armoured cruiser *Dupuy de Lôme*, and the United States cruiser known as No. 12, of 21 knots speed and 20,000 I.H.P., and some Italian torpedo cruisers engined by Messrs. Hawthorn, Leslie, & Co.

There would appear to be several advantages to be anticipated from the adoption of triple screws in high-speed ships of war. It is possible by this means to effect a reduction in weight of machinery since a higher speed of revolution is admissible. There would probably be a saving of fuel when cruising at slow speeds by using the centre screw alone. In twin screw ships of small beam in proportion to length, it is more economical to use one propeller only at slow speeds, and to disconnect the other, as the drag of the idle screw and of the small angle of helm required to maintain a course, are more than compensated for by the reduction of engine friction and the superior economy of steam obtained by working one engine at a greater proportion of its maximum power, and a still further economy might be expected from the increased subdivision of engine power in triple screw ships.

* Trans. Inst. Naval Architects, xxvii. p. 239.

CHAPTER IV.

EXPERIMENTS WITH MODELS AND THEIR APPLICATION TO THE DETERMINATION OF THE MOST SUITABLE DIMENSIONS.

For a screw of any given pitch-ratio, there is a particular slip-ratio corresponding to its maximum efficiency. A greater or less amount of slip than this will result in a smaller return of useful work in proportion to the power expended in driving the screw. By slip-ratio is meant the ratio P R to V, where

P = mean pitch;
R = revolutions;
V = velocity of feed, or speed of screw through the water.

The best way of determining the slip-ratio which is suitable for any given ratio of pitch to diameter is by experimenting upon a series of model screws of some selected type, each differing only in the ratio of pitch to diameter, and the following conditions may be laid down as essential, if the results obtained are to be useful for general application :—

1. Each model must be tried at a number of different slip-ratios.

2. The velocity of feed must be capable of accurate measurement.

3. The power expended in driving the screw must be measured, and it must be the power put into the screw-shaft, and not complicated with engine friction, which is an unknown quantity.

It is clear, therefore, that no experiments would be satisfactory, in which the screw under examination was working in the wake of a vessel, because it would then be impossible to measure the velocity of feed, since the forward motion of the wake is an unknown quantity, and varies with the speed of the ship in an unknown manner.

The late Mr. W. Froude, by means of an ideal conception of a small element of helical surface, rotating at the end of a non-resisting radial arm, deduced by theory for the screw, results very similar to those which were afterwards yielded by experiment.* In 1877, Mr. Froude described how such experiments were being conducted at Torquay,† and a very full account of the system pursued, not only for ascertaining the screw efficiency, but also for investigating the effect upon the operation of the screw of the presence in front of it of the hull of the ship, was given by Mr. R. E. Froude in 1883.‡ All these papers should be consulted by any one proposing to experiment for themselves.

* Trans. Inst. Naval Architects, xix. p. 47.
† Proc. Inst. Civil Engineers, li. p. 38.
‡ Trans. Inst. Naval Architects, xxiv. p. 231.

In the years 1879–80, Mr. John I. Thornycroft made a number of experiments with models of small dimensions, and a detailed description of these will be given as they are interesting as showing what can be done with moderately simple appliances.

The models were about 9 inches in diameter, and this size was found to be convenient. The maximum thrusts did not exceed 30 lbs., so that although the scale was large enough to admit of accurate measurement, a moderately small dynamometric apparatus could be employed. They were made as follows:—A wooden block was prepared from the reduced propeller drawing, upon which a blade was modelled by hand in paraffin. A mould of the blade was made in plaster of paris, into which was run an alloy consisting of tin and bismuth, the latter in small proportions. This material is sufficiently soft to be scraped and cut with a knife, and at the same time is strong enough to retain its form. It, of course, does not rust. The cast blade was then filed, burnished, and accurately fitted to the wooden block if it had become at all distorted. The blades were secured in the boss by screws in such a way that the pitch could be varied to any desired extent. A steam launch was fitted up with a small shaft passing through the bow to carry the model screw, the shaft projecting a sufficient distance in front of the launch to ensure that the model should work in

undisturbed water. This shaft could move very freely in its bearings to and fro, and the end of it was attached by means of a steel pianoforte wire to a spring, so that the thrust exerted by the propeller could be recorded. This shaft was made to revolve by means of a gutband working on to a pulley, and driven by means of a small engine of one or two horsepower. The measurements made were:—

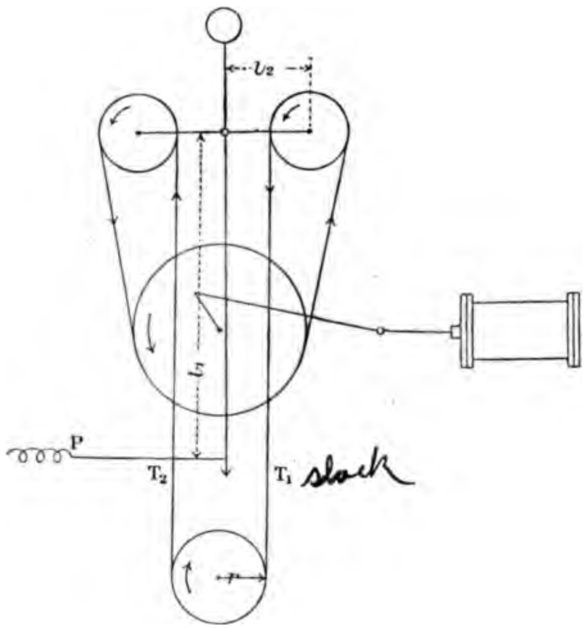
1. The thrust exerted by the model.
2. The revolutions of the model.
3. The speed of the launch.
4. The turning effort expended in driving the model.
5. Equal intervals of time.

The constant friction of the engine and shafting was also measured in order to get the true zero for the turning effort diagram. A dynamometer was constructed, by which records were continuously made upon a revolving drum driven at a uniform speed by means of clockwork. A number of pens over the drum were each connected to an electromagnet in such a way, that so long as no current flowed the pens were stationary, and traced straight lines upon the paper as it revolved beneath them. When contact was made the pens were jerked and made a lateral indent in the line. One pen was electrically connected with a clock, and measured intervals of time, making an indent every 12 seconds. A second pen recorded the revolutions

of the main engine driving the launch, these revolutions affording a means of checking the speed. A third pen recorded the revolutions of the model, a counter on the shaft making contact every 50 revolutions. The speed of the launch was found by passing a fixed distance on shore of 300 feet, the time of passing the posts being marked by an electric pen actuated by the observer pressing a button. As the observations were taken in a tideway two runs were necessary to determine the speed for every observation, one upstream and one down. Another pen was connected to a spring before mentioned, to which the model shaft was attached, and recorded the extension of the spring and thrust of the propeller. The last pen showed the tension of the gut driving the model, and thus measured the turning effort upon the shaft. This tension was obtained by the arrangement shown in Fig. 26. The large pulley in the centre is driven by the small engine of which the cylinder, piston-rod, connecting rod, and crank-arm are indicated in the figure. The lower pulley is on the shaft of the model propeller. The two upper pulleys are carried by a bar pivoted at the centre. Rigidly attached at right angles to this bar is a long lever, the weight of which is balanced by the ball at the top. The motion of the lever is limited to a short travel on each side of the vertical by stops not shown in the figure. A spring is attached to the bottom of the lever, and its exten-

sion is automatically recorded upon the diagram. The driving band is passed round the pulleys as shown, the direction of its motion being indicated by arrows. When the central pulley is made to

FIG. 26.



revolve, the tension of the gut pulls down the left-hand pulley, and extends the spring until its tension is sufficient to prevent further motion of the lever. An adjustment is provided in the cord between the spring and the lever, so that the latter may be maintained approximately vertical.

If T_2 is the tension of the ascending or tight side of the band, T_1 the tension of the descending

or slack side, and P the pull in pounds as measured by the spring, then

$$T_2 - T_1 = \frac{P \times l_1}{2\,l_2},$$

and turning moment $= (T_2 - T_1) \times 2\,\pi\,r \times$ revolutions of r per minute.

The launch, which as before stated was driven by an independent screw, maintained an approximately constant speed of about $4\frac{1}{2}$ knots, and a number of observations were taken at different revolutions of the model and plotted as shown in Plate 1. Curve A is the thrust of the model, B is the useful work in foot-pounds per minute, being the product of the thrust into the speed through the water. C is the work expended in foot-pounds per minute. The useful work divided by the work expended is a measure of the efficiency of the model as shown by curve D.

A convenient way of utilising the results thus obtained is to construct a series of constants, which will express the relation between disc-area, power, and speed, at different slip-ratios. A second series of constants can be formed expressing the relation between diameter, speed, and revolutions. These constants depend upon the following laws:—

1. For a given pitch-ratio and efficiency the disc-area is proportional to the horse-power, and inversely proportional to the cube of the speed.

2. For a given pitch-ratio and efficiency the revolutions per minute are proportional to the

speed, and inversely proportional to the diameter. They might take the following forms :—

$$C'_A = \text{disc-area in square feet} \times \frac{v^3}{\text{H.P.}} ;$$

$$C_B = \text{revolutions per minute} \times \frac{D}{v} .$$

Where v = velocity of feed;
H.P. = effective horsepower in the screw-shaft.

In this shape, however, it would only be possible to obtain from them directly the proportions proper for a screw to propel a "phantom ship"—that is a ship which would require the same thrust to propel it at any given speed as a real ship, but which will create no disturbance in the water, driven by a "phantom engine"—that is an engine without friction. In order to make them available for general use, it is more convenient to substitute V = speed of ship for v = velocity of feed, and I.H.P. for effective horsepower in the shaft. In order to do this it is necessary to make certain assumptions as to the speed of the following current and as to the ratio $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ and as these will vary with the form of the ship and with the type of engine respectively, an element of uncertainty is here met with, and much will depend upon the judgment of the designer, as to whether it is necessary to apply a wake correction or a propulsive coefficient correction, or whether the standard values assumed for these factors may be supposed to be a sufficiently close approximation.

The standard wake has been taken as 10 per cent. of the speed of the vessel. In a very full ship it might be as much as 30 per cent. Therefore V the speed of the ship should be reduced, when using the constants, by 20 per cent. for a very full ship, and by amounts varying from 20 per cent. to nil, as the fulness of form varies from "very full" down to what may be considered a "fairly fine" vessel when no correction need be made.

Table II. p. 74 gives the value of the wake corrections for a few vessels. The standard ratio of $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ has been taken as ·5. A correction can be made for any deviation from this assumed value. If, for example, the E.H.P. is estimated at 55 per cent. of the I.H.P., the I.H.P. must be multiplied by the ratio $\frac{55}{50}$.

Again, the constants are primarily correct for four-bladed screws; they can be used for three-bladed or two-bladed screws by multiplying the I.H.P. by $\frac{1}{0\cdot865}$ or $\frac{1}{0\cdot65}$ respectively.

The form, therefore, which the constants finally take is

$$C_A = \text{disc-area in square feet} \times \frac{V^3}{\text{I.H.P.}};$$

$$C_R = \text{revolutions per minute} \times \frac{D}{V}.$$

Where V = speed of ship in knots.

If constants are constructed in this way from the curves in Plate 1 corresponding to different amounts of slip, a row of figures is obtained, such as is shown by any one of the horizontal lines in Table I., p. 73, and these can be placed in their proper relative position under a curve of efficiency. For example, to find the constants C_A and C_B corresponding to 750 revolutions of a three-bladed model from the curves in Plate 1 :—

Diameter of model	= 9 inches.
∴ Disc-area	= 0·441 sq. feet.
Work expended in foot-pounds	= 9625.

$$\text{I.H.P.} = \frac{9625}{33{,}000} \times 2 = \cdot 583.$$

$$\frac{\text{Useful work}}{\text{Thrust in pounds}} = \frac{6330}{14 \cdot 8} = 426 = \left\{ \begin{array}{l} \text{velocity of feed in feet} \\ \text{per minute.} \end{array} \right.$$

$$\frac{426}{101} = 4 \cdot 22 \text{ knots.}$$

$$C_A = \frac{\cdot 441 \times \left(4 \cdot 22 \times \frac{1}{\cdot 9}\right)^3}{\cdot 583 \times \cdot 865} = 90.$$

$$C_B = \frac{750 \times \cdot 75}{4 \cdot 22 \times \frac{1}{\cdot 9}} = 120.$$

The object Mr. Thornycroft had in view in making the experiments was to test the efficiency of the screw-turbine propeller as compared with the screw he was then using on torpedo boats,* and he therefore did not carry out a complete set

* See a paper by Mr. Thornycroft in Trans. Inst. Naval Architects, xxiv. p. 42.

of experiments upon a common screw, which would have involved a series of trials with a number of models of varying pitch-ratio, but contented himself with ascertaining that a small change of pitch on either side of that for which the propeller was cast, obtained by twisting the blades in the boss, did not increase the efficiency, but rather reduced it.

This complete series was, however, afterwards carried out by Mr. R. E. Froude at Torquay, and the results were given by him in a paper read before the Institution of Naval Architects in 1886.* They corroborated generally those obtained at Chiswick, except in one particular. So far as could be judged by Mr. Froude, there was scarcely any difference in maximum efficiency within such a large range of pitch-ratio as from 1·2 to 2·2. As almost precisely the same maximum was obtained by Mr. Thornycroft with a pitch-ratio of 1·14, it seems reasonable to suppose that these even may hardly be considered as hard and fast lines beyond which efficiency will decline, and Mr. Froude considers that they may be fairly extended to 0·8 on the one side and 2·5 on the other.

It must, however, be borne in mind that this equality of efficiency is manifested by screws working in open water. There is a very general consensus of opinion that small pitch-ratios give

* Trans. Inst. Naval Architects, xxvii. p. 250.

the most favourable results in practice. If this opinion is justified, the explanation must be that large pitch-ratios cause a greater augmentation of hull resistance. That this would be so might be inferred from the reasoning on p. 21, which leads to the conclusion that such screws produce a greater suction than those of fine pitch. Although accepting the parity of efficiency of different pitch-ratios within the limits experimented upon, when the screws are considered apart from the vessels they propel, the author thinks that there is reason to suppose that there will be a loss involved in the use of a screw of coarse pitch, if it is placed in such a position that the increased suction produced by it is able to take effect upon the hull of the vessel.

Mr. Froude also found a great similarity between the curves of efficiency at different pitch-ratios, the only apparent effect of change of pitch-ratio being to cause the maximum efficiency to occur at different slip-ratios. Where, for example, the efficiency of one propeller reached a maximum at 15 per cent. slip, the efficiency of another of different pitch-ratio was at a maximum at 20 per cent. slip, and so on. It was therefore possible to superimpose the curves and cause them to coincide by the simple device of empirically changing the scale of slip-ratio. Mr. Froude very kindly gave the author permission to give the results of the Torquay experiments in a paper for the Institution

of Civil Engineers on "The Screw Propeller,* for which purpose the author compiled Table I., p. 73, in which each horizontal line of figures corresponds to a particular pitch-ratio and contains constants for disc-area, and revolutions at different amounts of slip as calculated from a set of curves such as that shown in Plate 1, and in the manner already described. These all occupy their proper relative positions under the curve of efficiency.

The table embraces the whole of the experiments possible with a particular type of screw, including pitch-ratios extending from 0·8 to 2·5, and slip-ratios from the lowest to the highest which is considered practicable.

It would be used in the following manner:— Let it be supposed, for example, that the size of the screw is limited by the draught of water. If the given disc-area is multiplied by the cube of the speed of the vessel in knots and divided by the I.H.P., the constant C_A is obtained. Suppose it is 360. The nearest figure to this in the column under the maximum efficiency should be sought, and its position, when found, indicates the pitch-ratio 1·6, which will be in the same line at the left hand of the table. Adjoining the disc-area constant 360 will be found the revolutions constant 71. This number, multiplied by the speed of the vessel in knots, and divided by the diameter of the screw in

* Proc. Inst. Civil Engineers, cii. p. 74.

feet, will give the number of revolutions at which a four-bladed screw should run to obtain the maximum efficiency.

It is evidently desirable to select the constants from the column under the maximum efficiency, but in special cases when the revolutions are required to be either exceptionally high or exceptionally low in order to suit existing engines, the same disc-area constant may be taken from one of the other columns where it will be found associated with either a lower or a higher value of C_R accordingly as the slip ratio is greater or less; and it is possible to see at a glance what sacrifice it is necessary to make in efficiency in order to obtain the required result.

If the product of the C_R constant multiplied by its proper pitch-ratio is greater than $101 \cdot 3\dot{3}$ the apparent slip will be positive; if less, it will be negative. The amount of the slip in either case will be given by

$$\text{Slip per cent} = \frac{p\,C_R - 101 \cdot 3\dot{3}}{p\,C_R} \times 100;$$

where p = pitch-ratio.

The same constants are presented in a graphic form in Plate 2, in which each vertical column of Table I. is plotted as a curve and values of C_A and C_R corresponding to intermediate pitch-ratios may be thus obtained.

The method of correcting for two and three blades and also for different values of wake is

due to Mr. R. E. Froude, and Table II. was given by him for the purpose of fixing upon a suitable wake correction in his masterly paper already referred to,* which is worthy of the most careful study. Mr. Froude's screws were of uniform pitch, and the blades were elliptical. The width in the middle of the developed blade was $0 \cdot 4 \frac{D}{2}$.

It follows that the developed surface, supposing each blade to be a complete ellipse, would be

For a four-bladed screw	=	disc-area	× 0·4
„ three „	=	„	× 0·3
„ two „	=	„	× 0·2

As the developed area is usually taken as exclusive of the boss, the portions of the ellipses cut off by it must be deducted. A few examples of the use of the tables will be given at the end of the chapter.

It will be noticed that the disc-area constants in the columns under maximum efficiency permit great latitude in the choice of diameter for a given I.H.P. and speed, so that if consideration is confined to the screws alone apart from the vessels they are designed to propel and the service these vessels are intended to perform, within these limits the efficiency is independent of the actual size. For example, take the case of a vessel of

* Trans. Inst. Naval Architects, xxvii. p. 250.

good form having engines of 500 I.H.P. and expected to attain a speed of 10 knots. From Table I. equal efficiency may be expected with a screw having a diameter of 10 feet and 0·8 pitch-ratio, and with one having a diameter of 15½ feet and a pitch-ratio of 2·5. The first would run at 138 revolutions per minute, the second at 33½. Some remarks have already been made (see p. 61) touching the effect of pitch-ratio upon efficiency, but in considering the relative advantages of large and small screws the duties required of the vessel must be taken into account. Suppose it is desired, for example, to maintain a high speed against head winds and seas. It is generally supposed that a large screw or a large surface is all that is required, but if we consider what happens when a vessel meets head winds we shall see that this is not necessarily so. In such circumstances the speed of the ship is checked, the revolutions of the screw remaining practically unaffected, so that the slip-ratio is increased. If this is already sufficient to give maximum efficiency at the smooth water speed, the efficiency will be reduced when the slip is increased by the wind, and it is probable that the 15½-foot screw of 2·5 pitch-ratio would waste as much power as the 10-foot screw of 0·8 pitch-ratio, although one has nearly 2½ times the surface of the other, because they both have the same position to start with as regards the curve of efficiency. Large diameter

associated with large pitch-ratio is valueless for the purpose; what is required is that the slip-ratio shall not be excessive when the speed of the vessel is retarded by external resistances. The case is analogous to that of a tug, and must be similarly treated. The best proportions will be obtained by designing for a speed less than the maximum smooth-water speed, but such as the vessel is expected to maintain over an average passage. The propeller would have a somewhat reduced efficiency when the vessel was developing her full power over the measured mile, would in fact be too large, but would work at its best at the speed assumed as the average, and should effect a saving of fuel on the voyage. When the diameter is limited the blade-surface may be increased with advantage.

It has been stated by Mr. Hall-Brown * that for vessels of very full form—a class with which he has had much experience—a large diameter and a small pitch-ratio are essential to success, and he attributes the necessity to the influence of dead water as distinguished from frictional wake (see p. 33). In such a case the blades must reach well out into water clear of the stern, so that the proportion of dead water to the total area of stream acted upon may be as small as possible. Mr. Hall-Brown gives the particulars of what is found to

* Proc. Inst. Civil Engineers, cii. p. 131.

be a good screw for a cargo vessel of the following dimensions :—

Length B.P.	277 feet
Beam (moulded)	37·5 „
Draught	19 feet 11 inches
Displacement	4670 tons
Block coefficient	·792
I.H.P.	825
Speed	9 knots
Diameter of screw	16 feet
Pitch „	16 „
Revolutions	64

The values of C_A and C_B are 184 and 115 respectively, which agree with the constants in the table for 1·0 pitch-ratio at 69 per cent. efficiency, but as the table is calculated for a wake value of 10 per cent., corresponding to a fine form of vessel, he suggests that this points to the conclusion that the propulsive coefficient is very low on account of the action of the screw upon the dead water, and that the two corrections for wake and propulsive coefficient tend to annul one another. In such a case it might be expected that a better performance would be obtained if twin screws were employed, as these would be to a great extent clear of the dead water.

Whenever the ship is of exceptional form, no exact rules can be given for the proportions of screws deduced from model trials in undisturbed water. Certainty can only be obtained by trying the model screw behind a model of the ship, and this is always done by Mr. Froude in the case of

new Admiralty designs. But every carefully recorded ship trial is in one sense a model experiment of this character, and when a screw is to be designed for a ship of a special type, it is safer to calculate the values of C_A and C_R from the actual figures obtained from the trials of some vessel, of proportions as similar as possible, which has given a good result, because the values of wake and propulsive coefficient may then be assumed to be similar also. In place of constructing constants, the following more direct method may be employed :—

To find the diameter of a propeller for a given I.H.P. and a given speed from the diameter of another similar propeller at a different I.H.P. and a different speed.

If d = diameter of model, which may be larger or smaller than D;
D = diameter of required propeller;
p = I.H.P. of model;
P = I.H.P. of required propeller;
v = speed of vessel with model propeller;
V = " " required "
r = revolutions of model propeller;
R = " required "

Then

$$D = \sqrt{d^2 \times \frac{v^3}{V^3} \times \frac{P}{p}}$$

and

$$R = r \times \frac{V}{v} \times \frac{d}{D}.$$

The pitch-ratio must be the same as that of the screw which is treated as the model.

Examples in the use of Tables I. and II.

Example 1.—Find the diameter and revolutions of a screw to work at maximum efficiency for a vessel of 20 knots speed and 6000 I.H.P. Pitch-ratio to be 1·2.

The disc-area constant (C_A) in the table for this pitch-ratio is 288.

The revolutions constant (C_R) in the table for this pitch-ratio is 92.

$$\text{Disc-area} = C_A \times \frac{\text{I.H.P.}}{(\text{Speed in knots})^3} = 288 \times \frac{6000}{20^3} = 216 \text{ sq. ft.}$$

$$\therefore \text{ Diameter} = 16\cdot5 \text{ feet.}$$

$$\text{Revolutions} = C_R \times \frac{\text{speed in knots}}{\text{diameter in feet}} = 92 \times \frac{20}{16\cdot5} = 111.$$

Example 2.—Find the pitch and revolutions of a screw to work at maximum efficiency for a vessel of 20 knots speed and 6000 I.H.P. Diameter not to exceed 15·5 feet.

$$\text{Disc-area} = 189 \text{ square feet.}$$

$$C_A = 189 \times \frac{20^3}{6000} = 252.$$

Nearest disc-area constant in table under maximum efficiency is 251 at pitch-ratio 1·0.

$$\therefore \text{ Pitch} = 15\cdot5 \text{ feet.}$$

The corresponding value of C_R is 109.

$$\therefore \text{ Revolutions} = 109 \times \frac{20}{15\cdot5} = 141.$$

Example 3.—Find the pitch-ratio and efficiency of a screw for a vessel of 20 knots speed and 6000

I.H.P. The diameter to be 15·5 feet and the revolutions about 80 per minute.

Disc-area = 189 feet.

$$C_A = 189 \times \frac{20^3}{6000} = 252.$$

$$C_B = 80 \times \frac{15 \cdot 5}{20} = 62.$$

The nearest constants in table are at pitch-ratio 2·2 and efficiency 68 per cent. Where the diameter and revolutions are both limited, the curves on Plate 3 will probably be found more convenient, as intermediate pitch-ratios can be selected.

Example 4.—Find the diameter and pitch of a screw to work at maximum efficiency for a vessel of 20 knots speed and 6000 I.H.P. Revolutions to be 85. Wake correction to be made for a form of the fulness of H.M.S. *Devastation*, corresponding to a wake percentage of 15·8.

The multiplier from Table II. is 0·942.

$$20 \times 0 \cdot 942 = 18 \cdot 8 \text{ knots.}$$

By trial and error it will be readily found that the constants 306 and 85 for disc-area and revolutions respectively, at 1·3 pitch-ratio, will give the required number of revolutions, thus :—

$$306 \times \frac{6000}{(18 \cdot 8)^3} = 276 \text{ square feet.} \quad \therefore D = 18 \cdot 75 \text{ feet}$$

and

$$85 \times \frac{18 \cdot 8}{18 \cdot 75} = 85 \text{ revolutions nearly.}$$

Example 5.—Find the diameter, pitch, and revolutions of a three-bladed screw to work at maximum efficiency for a vessel of 20 knots speed and 6000 I.H.P. Pitch-ratio to be 1·2.

$$C_A = 288. \qquad C_B = 92.$$

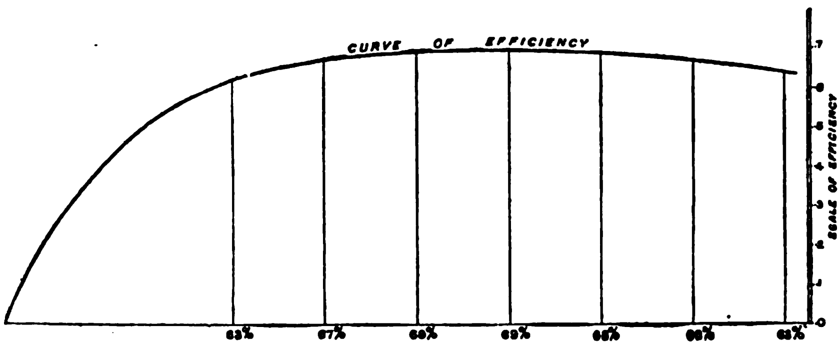
$$6000 \text{ I.H.P.} \times \frac{1}{0 \cdot 865} = 6940.$$

$$288 \times \frac{6940}{20^3} = 250 \text{ square feet.} \quad \therefore D = 17 \cdot 8 \text{ feet.}$$

$$92 \times \frac{20}{17 \cdot 8} = 103 \text{ revolutions.}$$

$$\text{Pitch} = 17 \cdot 8 \times 1 \cdot 2 = 21 \cdot 3 \text{ feet.}$$

Table I.

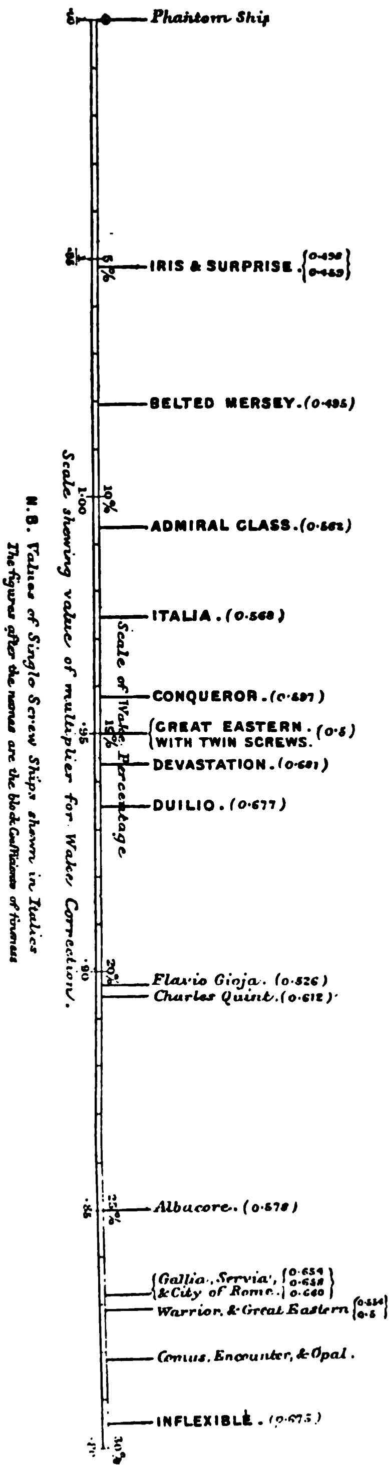


Pitch-ratio.	C_A	C_R	C_A	C_R	C_A	C_R	C_A	C_R	C_A	C_R	C_A	C_R	C_A	C_R
0·80	468	122	304	128	215	134	157	142	115	150	86	160	65	171
0·90	506	109	329	114	234	120	170	127	125	135	93	144	71	154
1·00	546	99	355	104	251	109	184	115	135	123	100	131	76	140
1·10	585	91	380	95	270	100	196	105	144	113	107	120	82	128
1·20	625	83	405	87	288	92	210	97	154	104	115	111	87	119
1·30	665	77	431	81	306	85	224	91	163	97	122	103	93	111
1·40	704	72	456	76	325	80	236	85	173	90	129	97	98	104
1·50	742	67	482	71	342	75	250	79	183	85	136	91	104	98
1·60	780	63	507	67	360	71	263	75	193	80	144	87	109	93
1·70	..	..	533	63	378	67	276	71	202	76	151	82	115	88
1·80	..	..	558	60	396	64	290	68	212	73	159	78	120	84
1·90	..	..	584	57	415	61	304	65	222	69	166	75	125	81
2·00	..	..	609	55	432	58	315	62	231	67	173	72	131	77
2·10	..	..	635	52	450	56	329	59	241	64	180	69	136	75
2·20	..	..	660	50	469	54	342	57	250	62	187	67	142	72
2·30	..	..	685	48	486	52	355	55	260	59	194	64	148	69
2·40	..	..	710	47	505	50	369	53	270	57	202	62	153	67
2·50	..	..	736	45	523	48	381	52	280	56	209	60	159	65
	5		7		9		11		13		15		17	

Scale of Abscissa value.

$$\text{Disc-area} = C_A \times \frac{\text{I.H.P.}}{(\text{Speed in knots})^3}. \qquad \text{Revolutions} = C_R \times \frac{\text{Speed in knots}}{\text{Diameter in feet}}.$$

TABLE II.


Phantom Ship
IRIS & SURPRISE. (0·436) (0·469)
BELTED MERSEY. (0·495)
ADMIRAL CLASS. (0·562)
ITALIA. (0·568)
CONQUEROR. (0·597)
GREAT EASTERN WITH TWIN SCREWS. (0·5)
DEVASTATION. (0·601)
DUILIO. (0·677)
Flavio Gioja. (0·526)
Charles Quint. (0·612)
Albacore. (0·578)
Gallia, Servia, & City of Rome. 0·654 0·658 0·640
Warrior, & Great Eastern 0·554 0·5
Comus, Encounter, & Opal.
INFLEXIBLE. (0·675)
1·00
·95
·90
·85
5%
10%
15%
20%
25%
30%
Scale of Wake Percentage
Scale showing value of multiplier for Wake Correction.
N.B. Values of Single Screw Ships shown in Italics
The figures after the names are the block Coefficients of fineness

CHAPTER V.

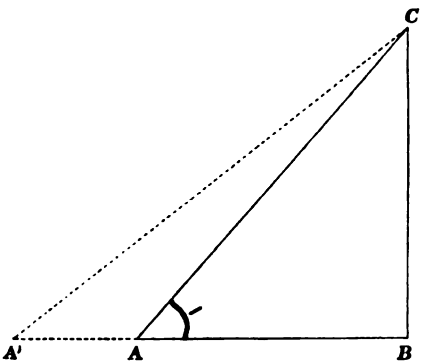
GEOMETRY OF THE SCREW.

If a point move on the surface of a cylinder in such a way that, while moving uniformly around the cylinder it advances uniformly in the direction of its axis, it will trace a curve known as the helix. Imagine the cylinder cut on one side by a straight line parallel to the axis meeting the helix in consecutive points A C (Fig. 27), and then unrolled and laid flat, the circumference through A will become the straight line A B; B C at right-angles to it will represent the direction of the axis while that part of the helix formed during a complete revolution of the tracing point will be represented by the line A C. Since the distances moved by the point in the directions A B, B C are proportional, A C is a straight line. B C being the distance moved through in the direction of the axis, while the point goes once entirely round the cylinder, is the pitch, while the angle B A C, which the unrolled helix makes with a plane at right-angles to the axis, is the angle of the helix or screw.

If a straight line move uniformly round an axis which it intersects, and to which it is always at

right angles, advancing at the same time uniformly in the direction of the axis, it will sweep out a surface known as a helicoid, and every point in the generating line will trace a helix as described

FIG. 27.



above, necessarily lying on this helicoid. Since during a complete revolution of the generating line every point moves through the same distance in the direction of the axis, the helicoid is a surface of uniform pitch, that is B C, Fig. 27, is constant for the helices traced by all points in the generating line. A helicoid can therefore be, and often is, used for the acting face of a screw-blade of uniform pitch. It is not necessary, however, that the generating line should be at right angles to the axis; such a surface may be generated by any line, straight or curved, moving uniformly along and revolving uniformly around an axis, intersecting and always making the same angle with it.

The helix traced by any point in the gene-

rating line will also be the curve of intersection with the screw surface of a co-axial cylinder of radius equal to the perpendicular distance of the point from the axis. The larger the radius of the cylinder, the larger of course the length of the circumference, as A′ B, Fig. 27, and as the pitch is constant, it follows that the angle of the helix must decrease as the radius of the intersecting cylinder increases. We thus arrive at the fundamental geometrical property of a surface of uniform pitch, viz. :—Co-axial cylinders intersect it in helices, all of which have the same pitch, but whose angles vary, decreasing as the radius of the cylinder increases. Near the axis, therefore, the helices will approximate in direction to that of the axis, and as the distance from the axis increases, they will lie more and more at right angles to it. If θ be the angle of the helix, p the pitch, and r the radius of the intersecting cylinder, $\tan \theta = \frac{p}{2 \pi r}$.

The curve of intersection of a co-axial cylinder with a screw surface will hereafter be referred to briefly as the "curve of intersection," or the "helix of intersection."

In commencing the design of a propeller for a given ship, the diameter and pitch must be first decided, and the considerations which will determine these having been fully dealt with in the preceding chapter it is only necessary to say that

it is usual to provide for an immersion of the tip of the upper blade equal to about $\frac{1}{10}$ the diameter of the propeller, and to allow a clearance from the ship's side varying from 6 inches in small ships to 1 foot in large ones.

With regard to the number of blades to be used, if the necessary disc-area can be obtained with a three-bladed propeller it is to be preferred, but if the draught of water is limited, a four-bladed propeller is practically equivalent to a three-bladed one of rather larger diameter.

The expanded blade-area of the Admiralty standard blade is an ellipse of major axis equal to the radius of the propeller, and of minor axis equal to $\frac{4}{10}$ the major axis. It is often found, however, that owing to the diameter being limited, sufficient blade-area is not obtainable by these proportions; in such cases the elliptical form is adhered to with an increased minor axis of from ·5 to ·55 the major.

If sufficient area cannot thus be obtained even with four blades, as in the case of shallow draught vessels where the diameter is very limited, the elliptical form may be greatly departed from, and the blade widened at the tip. A boss of diameter equal to $\frac{\text{diameter of propeller}}{3\frac{1}{4} \text{ to } 3\frac{1}{2}}$ will cut away about $\frac{1}{5}$ the area of the standard ellipse.

The expanded blade-area, which may be described as a flat surface of approximately equiva-

lent area to that of the blades, both as to amount and disposition, is derived as follows :—

A co-axial cylinder will intersect the screw surface in a helical curve making a certain angle with the axis, and it will intersect a plane passing through that diameter of the cylinder which passes through the middle point of the helical curve, and making the same angle with the axis in an elliptical arc. The length of the screw being small compared with the pitch, these two arcs will nearly coincide, and no great error will be involved by assuming that they do coincide. Imagine these elliptical arcs at all radii to be swung round a common centre line till they all lie in the same plane with their major and minor axes respectively coincident (though necessarily of different length), then the curve passing through their extremities will form the boundary of the expanded blade-area. This area is very nearly equal to the actual whole surface of the acting face of the blade, being in fact, somewhat less than it.

To draw a right-handed uniform pitch screw of Admiralty or Griffiths type, with elliptical blades of standard form, with generating line at right angles to the axis, with axis horizontal and centre line of blade upright :—

Draw a straight line A B, Fig. 28, Plate 3, to represent the axis of the propeller, and B C at right angles to it equal to the radius of the propeller. With B C as major axis and minor axis

equal to $\frac{4}{10}$ of B C, describe the ellipse B D C E. Draw the circle F H G of radius equal to that of the boss, cutting the ellipse in F G, the area G D C E F H G is the expanded area. Divide H C into a number of parts, preferably equal, at the points *a*, *b*, *c*, *d*, *e*, *f*. If p be the constant pitch of the propeller, take B A equal to $\frac{p}{2\pi}$ and join A H, A*a*, A*b*, A*c*, &c., these lines are called the pitch-lines. Then since the tangent of the angle which any one of these lines passing through a point on B C distant r from B, makes with B C is $\frac{p}{2\pi r}$, these angles are the angles of the corresponding helices of intersection, and, therefore, by the assumption previously explained, they are the angles which the planes of the elliptical arcs forming the expanded area make with the plane at right angles to the axis of the propeller when the arcs approximately coincide with the corresponding helices on the actual blade. Consider that elliptical arc on the expanded area passing through *c*, B*c* is its semi-minor axis, and since the angle *c* A B is its inclination when lying in its position on the actual blade to the axis of the intersecting cylinder, *c* A must be the length of its semi-major axis. Similarly A H, A*a*, A*b*, &c., are the lengths of the semi-major axes of the elliptical arcs through H, *a*, *b*, &c., respectively, and B H, B*a*, &c., are the corresponding semi-minor axes. It follows at once

that A is a focus of all these elliptical arcs, the other focus being at K where B K = B A. Draw, therefore, through the points H, a, b, &c., elliptical arcs with foci A, K and semi-major axes respectively equal to A H, Aa, Ab, &c. Then by the assumption, these elliptical arcs represent the helical arcs of intersection turned about B C till they lie in the same plane. If we reverse the process and turn them back from the expanded area through the same angle, their extremities will give us points on the outline of the blade. The angle any particular arc must be turned through is that which its pitch-line makes with B A.

This enables us to draw the projections, and in dealing with these we shall take a particular one of the elliptical arcs and show how to obtain the projections of the point at one of its extremities; the projections of the other extremity may be obtained in a similar way, but on the other side of the centre line. This will be a specimen arc. If the same method be adopted for the extremities of all the other arcs, series of points will be obtained, and if fair curves be drawn through the respective series, we have the required projections. For each projection of the blade we shall therefore deal only with the projection of one point on its outline.

Take any one of the elliptical arcs as the specimen, say that through c, viz. $c_1\, c\, c_2$, Fig. 28, Plate 3. Draw $c_2\, c_0$, parallel to the axis, mark off on the

pitch-line through c, $c\,c_3 = c_0\,c_2$. Draw $c_3\,c_4$ perpendicular to A B and $c\,c_4$ parallel to it. Then $c\,c_4$ is the projection of $c_0\,c_2$ on a fore and aft plane, and $c_3\,c_4$ is its projection on an athwartship plane. Set off $c_0\,c_5 = c_3\,c_4$, then c_5 is a point on the athwartship projection of the blade. Similarly on the other side of the arc we get another point c_6, and so on for all the other arcs.

It may be noticed that this projection may also be found thus: draw circular arcs with B as centre through H, a, b, c, &c. On the blade the elliptical arc approximately coinciding with the helical arc will project on an athwartship plane into a circular arc, the chord of the former of course projecting in that of the latter. Dealing with the arc through c, draw $c_0\,c_2$ parallel to the axis through c_2, meeting the circular arc in c_5, then the elliptical arc $c\,c_2$ will project on an athwartship plane into the circular arc $c\,c_5$. c_5 is therefore a point on the athwartship projection of the blade.

For the fore and aft projection take K P, Fig. 29, as the axis of the propeller, and draw P Q at right angles to it. Set off distances P H_7, Pa_7, Pb_7, &c., respectively, equal to B H_0, Ba_0, Bb_0, &c., and draw straight lines through the points H_7, a_7, b_7, &c., parallel to K P. Dealing with the line $c_9\,c_7\,c_8$, set off $c_7\,c_8$ equal to $c\,c_4$, Fig. 28, then c_8 is a point on the fore and aft projection of the blade. Similarly for other points.

For the horizontal projection take N R, Fig. 30, as the axis, R being forward, and draw straight lines through N making angles with N R equal to H A B, a A B, &c., Fig. 28, and inclining as shown since the blade is right-handed. These straight lines represent the directions on the actual blade of the chords of the elliptical arcs, the lengths are given in Fig. 28. Dealing with the line Nc', which is the direction of the chord $c_0 c_2$, Fig. 28, mark off $N\gamma = c_0 c_2$, Fig. 28, then γ is a point on the horizontal projection. On the other side of N we get similarly γ', the horizontal projection of the other extremity of the chord. Similarly for other points.

The complete projections of a three-bladed propeller are shown in Figs. 33, 34, 35, Plate 4. The athwartship projections of the two lower blades are simply repetitions of that of the upper blade, their centre lines being inclined to that of the upper one at angles of 120°, Fig. 33.

The fore and aft projection of the lower blade, Fig. 34, is obtained thus:—Across the projections of the blades in Fig. 33 and the top blade of Fig. 34 draw straight lines at right-angles to their centre lines as in Figs. 28 and 29. Consider the point whose athwartship projection is $\dot{c}_2$ on the right-hand lower blade, Fig. 33; this being on the following edge will be abaft the centre line in Fig. 34, the corresponding point on the leading edge (c_1) appearing on the forward side.

Then to obtain the position of this point on the fore and aft projection it must be borne in mind that it must lie in the same vertical plane perpendicular to the axis as it does when the blade is upright, and at the same distance from the horizontal plane through the axis as in its athwartship projection.

Consider the point whose athwartship projection is c_2, Fig. 33. Draw $c_3 c_4$, Fig. 34, on the left-hand side of the centre line P Q, parallel to the axis, at the same distance from it as c_2, Fig. 33, is from B P. Take $c_3 c_4$ equal to $c_0 c_7$ (c_7 being the position on the fore and aft projection which the point would occupy if the blade were upright), then c_4 is a point on the fore and aft projection of the right-hand lower blade. Similarly for other points.

Proceed in a similar manner for the other lower blade, bearing in mind that the upper edge, Fig. 33, is here the leading or forward edge. The projection of one blade only is shown in Fig. 34 to avoid confusion.

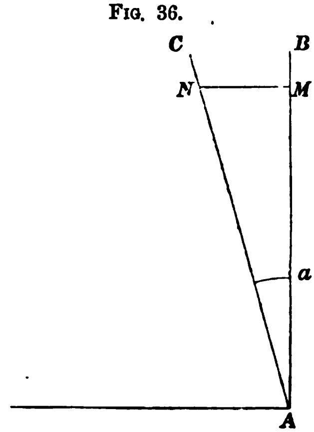
In the horizontal projection of the lower blades, Fig. 35, we proceed similarly. The point whose athwartship projection is c_2, Fig. 33, for example, will appear on the after side of the centre line of the blade at a distance from it equal to $c_0 c_7$, Fig. 34; and from the axis equal to the perpendicular distance of c_2, Fig. 33, from B E.

The projections of a three-bladed propeller are

thus completely determined. It may be remarked that it is often considered sufficient to replace the elliptical arcs on the expanded area, Fig. 28, by circular arcs with B as centre, forming the projections from the chords of these arcs precisely as has been described for the elliptical ones. The error introduced by this method of procedure is not great, being only appreciable towards the root of the blade, where it is of little consequence.

Blades are sometimes made with the generating line inclined to the axis, or in technical terms they are made with a skew. Let the two straight lines, A B, A C, Fig. 36, the former at right-angles to an axis, the latter inclined to A B at an angle a, moving together, generate screw surfaces of uniform and equal pitch. Then the helices of intersection of these two surfaces will be exactly similar, and one will be always a constant distance from the other, this distance being at a radius r, $r \tan a$. Imagine these two surfaces so far similar, that when A C at any radius leaves the surface, A B at the same radius leaves its surface, then the expanded area of the surface so formed by A B will represent what may be termed the effec-

FIG. 36.



tive expanded area of that formed by A C, and this should be of the elliptical or other form which would have been used if the generating line had been at right-angles to the axis. A blade generated by A C would therefore be formed from a blade generated by A B, simply by setting the helices of intersection, definite distances aft, the distance at M for example being M N. It follows therefore that the athwartship projection of the blade for the same "effective" expanded area is independent of the skew, and consequently for a skew blade with effective expanded area as for the blades shown in Figs. 33, 34, 35, the athwartship projection will be as in Fig. 33, no matter what the skew be.

The fore and aft projection of the upper blade, Fig. 37, will be formed by using a centre line inclined to the vertical at an angle equal to the inclination of the generating line to the vertical, and proceeding as in Fig. 29, setting off the distances horizontally. The projections of the lower blades are determined in a similar way to that described for Fig. 34. For example, the point whose athwartship projection is c_2, Fig. 33, will appear on Fig. 37 at c_4, lying in the same vertical straight line as c_7 (c_7 being the position of this point when the blade is upright), and being perpendicularly away from the axis, a distance equal to that of c_2 from the horizontal plane through the axis.

Next consider the horizontal projections, Fig. 38.

For the top blade the pitch-lines are not all drawn through the same point as in Fig. 35, but each line is drawn at the corresponding angle to the axis through a point on the axis at a distance from $M_1 N_1$ (corresponding to M N in Fig. 37), equal to the distance of the corresponding point on the generating line P Q from M N, Fig. 37, the process then being as in Fig. 30. For the lower blade we proceed as in Fig. 35, for instance, the point (c_2, Fig. 33) corresponding to c_7, Fig. 37, when the blade is upright, will appear in Fig. 38 at c_8 at a distance from the axis equal to the distance of c_2, Fig. 33, from the vertical plane through the axis and from $M_1 N_1$, equal to the distance of c_7 from M N, Fig. 37.

The projections of a three-bladed propeller with skew blades are shown in Figs. 33, 37, 38, except that the left-hand lower blade of Fig. 33 has not been shown in Fig. 37 to avoid confusion.

Where the generating line is curved, the method is now obvious.

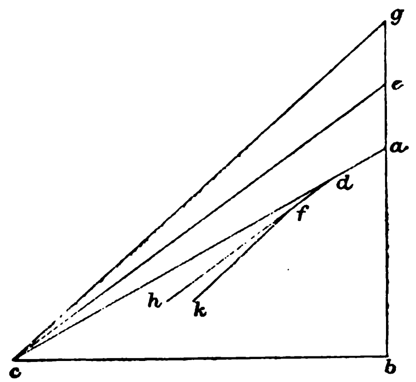
We pass on to consider blades not of uniform pitch.

If it is desired to make the pitch increase from the root of the blade towards the tip, that is, if the pitch at c, Fig. 28, for example, be 2π B Z instead of 2π B A, then the pitch-line at c will be Z c instead of A c. With this modification the method of constructing the projections is the same.

Blades are frequently made with an increasing

pitch from leading to following edge. In Fig. 39, let the pitch at the leading edge at a radius equal to $\frac{c\,b}{2\,\pi}$ be $a\,b$, and suppose as we go to a point d, the pitch increases to $b\,e$, then $e\,c\,b$ will be the pitch-angle at this point. Similarly if at f the

FIG. 39.



pitch increases to $b\,g$, then $b\,c\,g$ is the pitch-angle at f. Draw $d\,h$ parallel to $c\,e$, and $f\,k$ parallel to $c\,g$, then if the curve of intersection at the radius under consideration be unrolled as in Fig. 27, instead of being straight it will be as $a\,d\,f$—in other words, the pitch-line through a will not be straight; if the pitch varies continually, then the pitch-line will be a continuous curve.

A blade of uniformly varying pitch from leading to following edge would be generated by a line always intersecting, and always inclined at the same angle to the axis, moving uniformly

round the axis while advancing with uniform acceleration along it.

Only the acting face of the blade preserves the helical form, the back being made to give the required thickness at the different parts. The thickness at the centre line of the blade is first fixed at root and tip, and set off from the face of the blade. The two points thus found being joined, the distance of this line from the face gives the thickness at any intermediate point (see Fig. 29). The thickness at the tip should be as small as possible, consistent with good casting or forging, as the case may be, and in gun-metal is generally made about $\frac{3}{64}$ inch per foot of diameter. The blade when acting on the water is in the position of a beam under a load distributed all over its surface, varying in intensity, but it would be very difficult to find the bending moment at the root, and it is usual to make the rough assumptions that the total pressure on the blade tending to break it about the root section is proportional to the indicated thrust or to $\dfrac{P}{p\,R}$

(where P = I.H.P. per blade;
p = the pitch of the propeller;
R = the revolutions per minute),

also that the "leverage" is proportional to $D - d$, D being the diameter of the propeller, and d that of the boss; and that the moment of inertia of the root section of breadth b and depth h is propor-

tional to $b h^3$, then h the required thickness at the middle of the root is obtained from the formula

$$h^2 = \frac{c\,P.(D - d)}{p\,.\,R\,.\,b};$$

the value of the coefficient c being about 230 for gun-metal, and 90 to 100 for forged steel blades. (In the above formula, p, D, and d are to be taken in feet, and b and h in inches.)

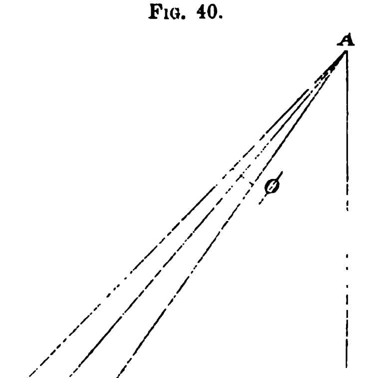
It is then usual to unroll the elliptical arc of Fig. 28 at any radius into a straight line, set off the thickness as found at its middle point, and describe a circular arc through the point so found and the extremities of the unrolled ellipse, and so obtain the thickness at other points than the middle (see Fig. 32).

When the blades are made separate from the boss they are usually attached to it by bolts arranged in a bolt circle of diameter C inches. As the thickness of the metal at the root is not disposed symmetrically about the centre of the flange, it is convenient to put one more bolt on the after side than on the forward side, and as the after bolts take the ahead load, which is greater than the astern, it is also an advantageous arrangement. If B be the combined area in square inches of the bolts at the smallest section (at the bottom of the thread) on the after side of the blade, h the distance of the centre of pressure of the blade from the under side of the

flange in feet, R and P being as before, then $B = \frac{P\,h\,K}{C\,R}$, the coefficient K ranging in value from 18 to 21 for gun-metal.

The number of bolts having been settled, their diameter is at once found. The bolts on the forward side are usually made of the same diameter for convenience. For the sake of possible adjustments that may be desirable on the trials of the machinery, and also for fining the pitch when it becomes necessary to reduce the steam pressure in the boilers, it is customary to make the bolt-holes in the blade-flange oval in order that the inclination of the blades to the axis may be altered. The amount of the oval is determined thus, Fig. 40.

FIG. 40.



Let the true pitch of the blade be $2\,\pi$ C E, and let the desired range of pitch be $2\,\pi$ E B and $2\,\pi$ E D respectively, on each side of this pitch. Take a

radius C A, so that A is about half-way up the blade. Join A D, A E, A B, then if ϕ be the angle D A B, the holes in the flange must be so elongated as to admit of the blade being turned through $\frac{\phi}{2}$ on each side.

The amount of the elongation on each side is therefore $\frac{\pi C}{360} \cdot \frac{\phi}{2} = \pi C \frac{\phi}{720}$, ϕ being measured in degrees.

If the axis of the helices of intersection of the blade be originally that of the propeller shaft, the acting surface will not be truly helical about the centre line of the shaft when the blade is turned round as just described, as the pitch is not altered uniformly. There are only two sections of the blade which receive the same change of pitch, and these are situated at the radii corresponding to a pitch angle of 45° in the case of the original and modified pitches respectively. Sections between these points receive a less change of pitch, and sections outside them a greater, in proportion to their distance from them. The effect produced therefore by twisting through any given angle depends upon the pitch-ratio; if this is small the critical points are near the boss, and twisting to augment pitch for example causes the pitch to increase throughout the greater part of the length of blade, the maximum occurring at the tip. If the pitch-ratio be such that the critical points fall

about the middle of the length, twisting to fine pitch will then result in a blade having the maximum pitch in the centre.

Figs. 29 and 31 show respectively longitudinal and transverse sections through the propeller boss.

CHAPTER VI.

THE HYDRAULIC PROPELLER.

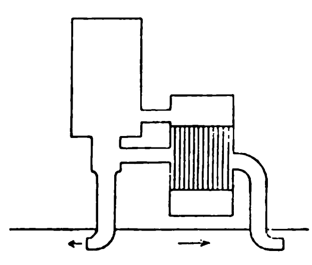
THERE are many reasons why a hydraulic propeller would be preferred to a screw or paddle in certain cases, if an economical result could be obtained with it, but the efficiency of the apparatus is necessarily so small that at the present moment it is believed that there is not a single vessel using the propeller for warlike or commercial purposes. When economy is a secondary consideration, and the circumstances are such as apparently to preclude the use of any propeller external to the vessel, the hydraulic propeller finds its opportunity. A steam lifeboat which has been recently built by Messrs. R. and H. Green of Blackwall for the National Lifeboat Institution, has been fitted with hydraulic machinery, made by Messrs. J. I. Thornycroft and Co., and she has met with a considerable measure of success. It was considered that the difficulty of keeping a screw immersed, and the danger of its becoming fouled by wreckage, or injured upon a sandbank, rendered it unsuitable, and justified the introduction of a propeller which could never race, and which was much less liable to injury.

The term jet-propeller, although in common use, is incorrect, the propeller in this system of propulsion consisting of a pump within the vessel, which discharges jets of water in a sternward direction, which are analogous to the race of a paddle or screw.

In 1879 a hydraulic vessel called the *Hydromotor* was built in Germany from the designs of Dr. Fleischer. In this vessel the engine and pump were combined, the arrangement being as follows :—

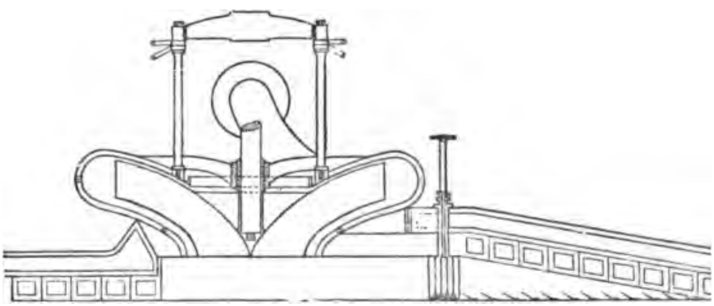
There was a cylinder lined inside with wood, at the bottom of which was a large pipe leading to a nozzle at the bottom of the vessel. A float of nearly the same diameter as the cylinder worked up and down in it. The cylinder being full of water and the float consequently at the top, steam was admitted by a valve above the float, and driving it down ejected the water through the nozzle Fig. 41. On reaching the bottom of its stroke, the float opened the exhaust, and the steam passed into the condenser. The vacuum then created in the cylinder caused the water to rise partly through the nozzle, but principally through a suction-valve in the bottom of the condenser. The cylinder was thus filled

FIG. 41.



with water, and the float rose to the top, in doing which it closed the exhaust and opened the steam-valve, and the operation was repeated. The loss by condensation appears to have been less than might have been expected in a cylinder filled alternately with steam and water, but as the cylinder was not entirely emptied at each stroke, a layer of boiling water always remained at the top and adhered to the wooden linings as the float descended. The information published is unreliable as no proper measured mile trials were made. All calculations made from indicated horsepower cards are of little value in this system, as the loss between the boiler and the indicator, which must be very large, is thereby

FIG. 42.

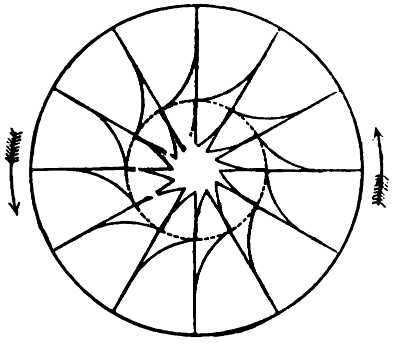


ignored. The only correct basis for comparison with either a screw or a turbine would be the amount of steam consumed per unit of work done.

In 1866, two armoured gunboats, the *Viper* and *Waterwitch*, were built, the former being propelled

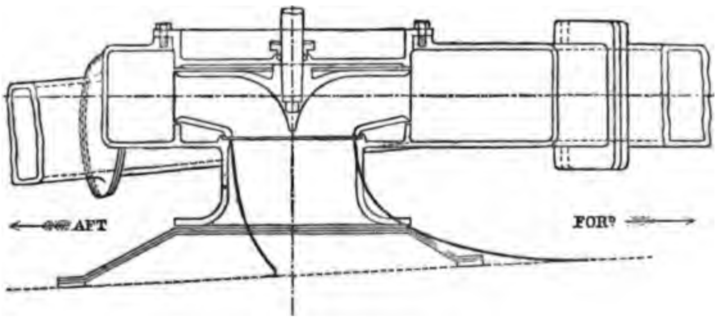
by twin screws, and the latter by hydraulic machinery, designed by Mr. Ruthven. The *Waterwitch's* propeller consisted of a turbine 14 feet indiameter, which drew water in at the

FIG. 43.



bottom of the vessel, and discharged it through two 24-inch nozzles at the sides level with the water (see Figs. 42 and 43).

FIG. 44.



In 1878, a hydraulic vessel was built by the Swedish Government for competition with a similar vessel with twin screws. The hydraulic vessel

was propelled by two turbines about 2 feet in diameter, which discharged water through submerged orifices at the sides near the extremities (see Figs. 44, 45, and 46).

Fig. 45.

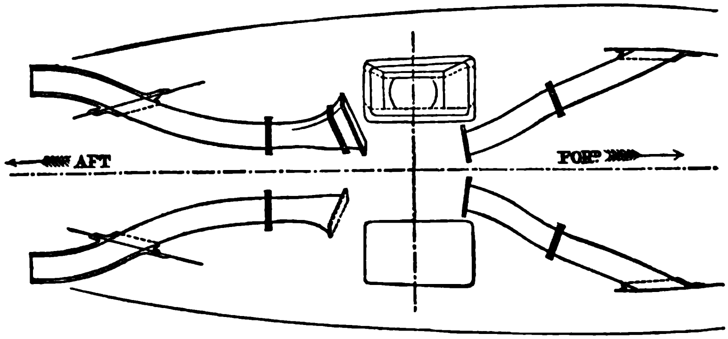
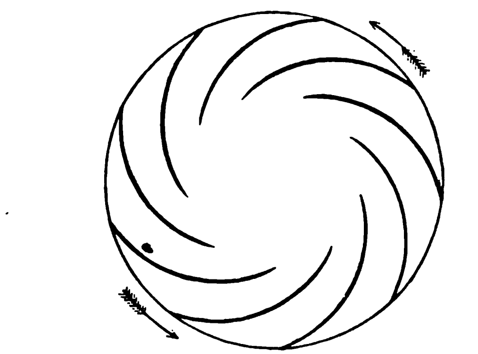


Fig. 46.



In 1883, Messrs. Thornycroft fitted one of a number of second class torpedo boats they were building for the British Admiralty with a hydraulic propeller consisting of a turbine 2 feet 6 inches in diameter, which discharged through two 9-inch

nozzles at the sides above water (see Figs. 47 and 48 and Plate 5). All these vessels were fully

FIG. 47.

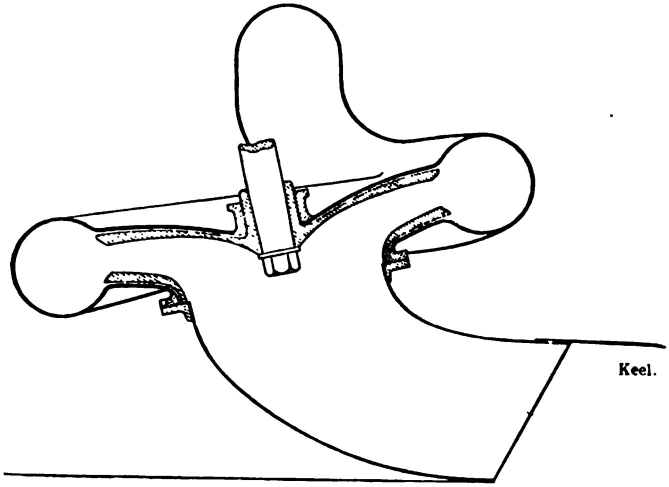
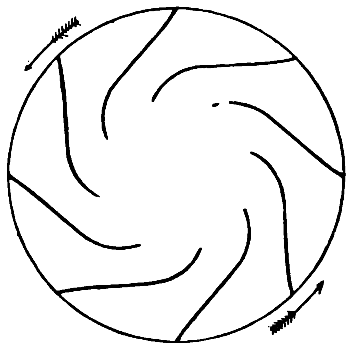


FIG. 48.



described by the author in a paper read before the Institution of Civil Engineers,* from which the

* Proc. Inst. Civil Engineers, lxxvii. p. 1.

figures and Tables III. and IV., p. 104, giving a detailed comparison of the performance of the respective screw and turbine vessels, are taken. In every case a very notable difference was found between them, and always to the disadvantage of the latter. There appears to be a loss of power corresponding to about 50 per cent. experienced by the hydraulic propeller as compared with the screw. The causes of this loss are not hard to find.

In the case of the *Waterwitch* and the Swedish boat, the water was received into the ship through a hole in the bottom in such a way as to suddenly arrest all the velocity which it had relatively to the vessel. In other words the entering water struck the ship and had the velocity of the ship impressed upon it before it entered the turbine. If the inlet is formed in the shape of a scoop, as was done in the Thornycroft boat (see Fig. 47) and Plate 5), and the water caused to change its direction gradually without having its velocity relatively to the ship checked, then this cause of loss is avoided. In such a case, if the vessel were towed along with the turbine removed and replaced by a curved channel connecting the inlets and outlets, the water would be scooped up, and would flow out at the nozzles, leaving them, if they are not above the surface, with a velocity relative to the ship equal to the speed of the ship, and with no velocity relative to still water except such as would be imparted by the friction of the passages.

The inlet of the Swedish vessel was subsequently altered as shown in thick black lines in Fig. 44, and the partial scoop thus formed caused an increase of speed from 7·87 knots to 8·12 knots, with the same expenditure of power.

Another loss of efficiency in the hydraulic system is due to the small area of stream acted upon, and the consequently high velocity which has to be imparted to it in order to give the necessary reaction (see p. 4). The reason why the area of stream acted upon is necessarily small, is that the size of the orifice which it is possible to make in a ship's bottom, is restricted by structural considerations, and must be very small indeed compared to the area of a screw's disc. Then again, the weight of water admitted into the ship is a serious consideration, as it represents so much loss of displacement.

A further waste of power is caused by the friction of the water in the pipes and passages, and by the changes in direction of its flow in passing through the bottom and out through the sides in a fore and aft direction. For these reasons the hydraulic propeller is essentially wasteful. In the screw and turbine competitive Thornycroft torpedo boats, the efficiencies were found to be as follows:—Screw boat: engine, 0·77; screw propeller, 0·65; total efficiency, 0·5. Hydraulic boat: engine, 0·77; pump, 0·46; jet, 0·71; total efficiency, 0·254.

The efficiencies of the pump and jet in this boat

were measured by the author in the following manner:—

A thin plate $1\frac{5}{16}$ inch square was attached to the end of a thin lever and placed in the jet just where it left the nozzle. The pressure on this plate was recorded by a dynamometer attached to the end of the lever. By finding the pressure upon a similar lever without the plate, the effect of the portion of the lever immersed in the jet could be allowed for. The apparatus was so arranged that the pressure could be measured at every point of the jet and not in the centre only. From the pressures on the plate the velocity of the stream at different parts of the jet was estimated, and from the mean velocity, the quantity of water discharged was calculated.

The relation between velocity of jet and pressure on the plate is as follows:—

$$\text{Pressure on plate} = \frac{0 \cdot 627 \times (\text{area}) \times \text{heaviness of fluid} \times (\text{velocity})^2}{\text{gravity}}.$$

If W = weight of water discharged per second;

V = speed of vessel in feet per second;

S = true slip or acceleration, or additional velocity impressed by the propelling apparatus in feet per second;

$V + S$ = velocity of discharge in feet per second;

g = acceleration produced by gravity in feet per second = $32 \cdot 2$;

Then the theoretical efficiency of the jet $= \dfrac{V}{V + \dfrac{S}{2}}$.

Efficiency of pump, supposing jet to have theoretical efficiency and the engine an efficiency of $0 \cdot 77$;

$$= \frac{\text{Work stored up in water}}{\text{Effective H.P. of engine}} = \frac{\dfrac{W V S}{g} + \dfrac{W S^2}{2g}}{\text{I.H.P.} \times 550 \times 0 \cdot 77}.$$

Efficiency of pump and jet

$$= \frac{\text{Useful work in jet}}{\text{Effective H.P of engine}} = \frac{\dfrac{W V S}{g}}{\text{I.H.P.} \times 550 \times 0 \cdot 77}.$$

Total efficiency

$$= \frac{\text{Useful work in jet}}{\text{Work expended}} = \frac{\dfrac{W V S}{g}}{\text{I.H.P.} \times 550}.$$

TABLE III.

—	Date.	Length.	Beam.	Maximum Draught.	Displacement.	I.H.P	Speed per Hour.	Midship Section.	$\frac{V^3 \times D^{\frac{2}{3}}}{I.H.P.}$	Number of Propellers.	Revolutions per minute.
		ft. in.	ft. in.	ft. in.	tons.		knots.	sq. ft.			
H.M.S. *Viper*	1867	162 0	32 0	11 10	1,180·00	696	9·58	337·0	141·4	2 screws	110
H.M.S. *Waterwitch* ..	1867	162 0	32 0	11 2	1,161·00	760	9·30	336·0	116·9	1 turbine	40
Swedish screw ..	1878	58 0	10 9	4 3	20·00	90	10·00	25·0	82·0	2 screws	250
„ hydraulic ..	1878	58 0	10 9	4 2	21·00	78	8·12	25·0	52·5	2 turbines	384
Thornycroft screw ..	1883	63 0	7 6	3 8½	12·89	170	17·30	11·9	169·0	1 screw	636
„ hydraulic	1883	66 4	7 6	2 6	14·40	167	12·60	13·4	72·0	1 turbine	428

TABLE IV.

—	Diameter of Turbine.	Area of Inlet.	Combined Area of Discharge.	Midship Section. / Area of Discharge.	Velocity of Discharge.	Water Discharged.	Efficiency of Pump and Jet.	Total Efficiency.
	ft. in.	sq. ft.	sq. ft.		ft. per sec.	lbs. per sec.		
H.M.S. *Waterwitch*	14 0	28·25	6·280	53·5	29·0	11,650	0·234	0·180
Swedish hydraulic	1 11½	1·62	0·864	30·5	28·0	1,510	0·277	0·214
Thornycroft hydraulic ..	2 6	1·52	0·951	14·1	37·2	2,210	0·330	0·254

CHAPTER VII.

THE SCREW-TURBINE PROPELLER.

This propeller was the fruit of the study given to the subject of hydraulic propulsion by Mr. Thornycroft, when designing the hydraulic torpedo boat already mentioned.

It has been pointed out in the preceding chapter that there are four characteristics of the centrifugal pump as applied for the purpose of propelling vessels, which prevent it from competing successfully with the paddle or the screw. These are:—

1. The difficulty of getting the water through the bottom of the vessel and into the pump without checking the velocity it already has relative to the vessel.

2. The necessity of carrying in the vessel all the water acted upon.

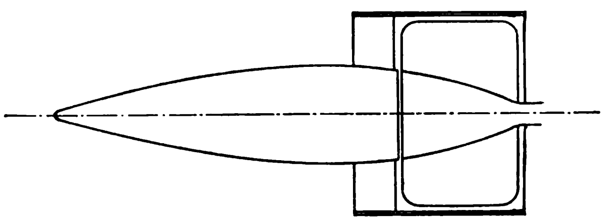
3. The loss caused by friction of the water in the pipes.

4. The loss due to bends in the passages.

It was obvious that if the turbine could be put outside the vessel and under the bottom, the first two causes of loss would be avoided, and also that if the water could be made to flow axially through

the turbine instead of radially as in the Ruthven pump, no pipes would be needed, there would be no loss by changes of direction, and the friction would be reduced to that due to passage through the turbine alone. The propeller illustrated in Fig. 49 was therefore devised upon these lines,

FIG. 49.



and as it is neither a screw nor a turbine strictly speaking, although allied to both, it has received the name of the screw-turbine.

It consists of a cylinder containing within it a body or boss of such a shape that the channel is gradually contracted from the forward to the after end. Within the forward part of the cylinder are placed revolving screw-blades attached to the forward part of the boss, which is in two portions.

The pitch of the forward edge of the screw-blades multiplied by the number of revolutions is approximately equal to the velocity of feed; the pitch increases uniformly along the length of the blade, imparting a uniform acceleration to the water. Aft of the revolving blades are placed numerous fixed blades of contrary curvature. The

area of the channel through the propeller is so proportioned as to suit the acceleration of the water caused by the blades. Thus at the forward end is a large opening which will admit a certain quantity of water at the velocity of feed; at the after end the area is restricted to that necessary to allow of the exit of the water at the speed of discharge. The long tapering body forming a prolongation of the boss outside of the cylinder, allows the streams of water to unite gradually without the formation of eddies. As the long pitch of the screw-blades causes considerable rotation of the water, the curved guides are so formed as to direct the water into a straight line aft, and the rotary motion is thus utilised without loss. When a model of this pro peller was experimented upon, the thrust of the revolving blades was measured separately from the thrust of the fixed guides. The latter was found to be quite considerable, amounting in the case of a very long pitch propeller to one-third of the total thrust.

The efficiency of the screw-turbine was found by experiment to be at least equal to that of the common screw, and a given thrust can be obtained with a much less diameter. It is therefore a very suitable propeller for vessels of shallow draught. As the water is not accelerated at all before it reaches the propeller, that is, as there is no sucking action, there would appear to be less augmentation of hull resistance caused by the screw-turbine than

by any form of open screw. It was stated on p. 21, that all open propellers work in a stream having a velocity varying between $V + S$ and $V + \frac{S}{2}$ depending upon the greater or less rotation of the race, the effect of the guide-blades in combination with the enclosed and contracted channel is to prevent rotation of the race, and to place the screw-turbine in the same condition as an open propeller would be in if it were possible to imagine one which, like Mr. Froude's "Acuator," produced no rotation. Its efficiency is therefore equal to $\frac{V}{V + \frac{S}{2}}$, and the loss of work is the least possible. Were it not for the large surface exposed to friction, it might be expected to have a much higher efficiency than the common screw.

It can be used with advantage, because of its relatively small diameter, in sea-going vessels, which often are in very light trim, and do not then properly immerse a common propeller, and therefore waste a large amount of power (see p. 34), and also in vessels in which the draught of water is so limited that the ordinary screw cannot be used. In such cases the screw-turbine possesses advantages as compared with paddle-wheels when high speed is required, because the weight of the machinery is much less than that of paddle engines of the same power, which necessarily run slowly.

The arrangement shown in Plate 6 has proved very successful when the draught of water is small. The launch illustrated has a draught of 12 inches. A tunnel is formed in the bottom of the boat, the top of which rises above the surface, the ends being submerged. A 16-inch screw-turbine is placed in the tunnel, so that one-fourth of the diameter of the propeller is above the water-level when the boat is at rest, but as soon as it moves, water is drawn up into the tunnel, and the air expelled by the action of the propeller, which then works completely submerged. There is no loss of power in lifting the water 4 inches above the level of the surface, because in falling it gives out the work expended in raising it. There is an incidental convenience in this arrangement. An air-tight door can be placed at the crown of the tunnel immediately over the propeller, which can be opened from inside the boat, since the admission of air to the tunnel causes the water within it to fall to the level of the outside water surface, and leaves the propeller partially emerged. It can then be examined and cleared if it should have become fouled, and if there are twin screws this operation can be performed upon one propeller while the other is revolving slowly.

Some vessels 140 feet by 21 feet, and having a draught of water of 1 foot 9 inches only, have been built upon this plan by Messrs. Thornycroft.

They were propelled by twin screw-turbines 32 inches diameter, and attained a speed of 15½ knots per hour. A launch 56 feet long and 15 inches draught of water has attained a speed of 16¼ knots with one screw-turbine 20 inches in diameter.

A smaller draught of water can be obtained by the use of this propeller than would be possible with the hydraulic propeller, on account of the excessive weight demanded by the latter for machinery and water.

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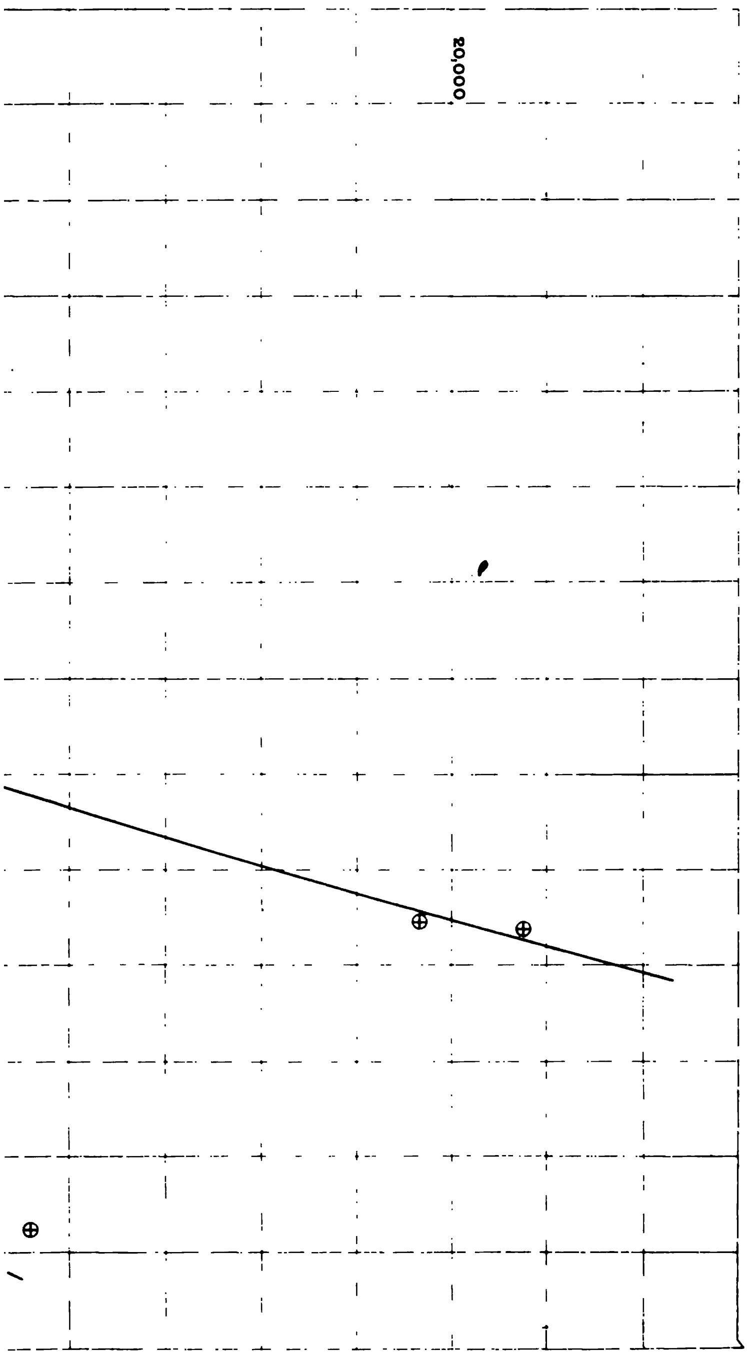
Plate 1.



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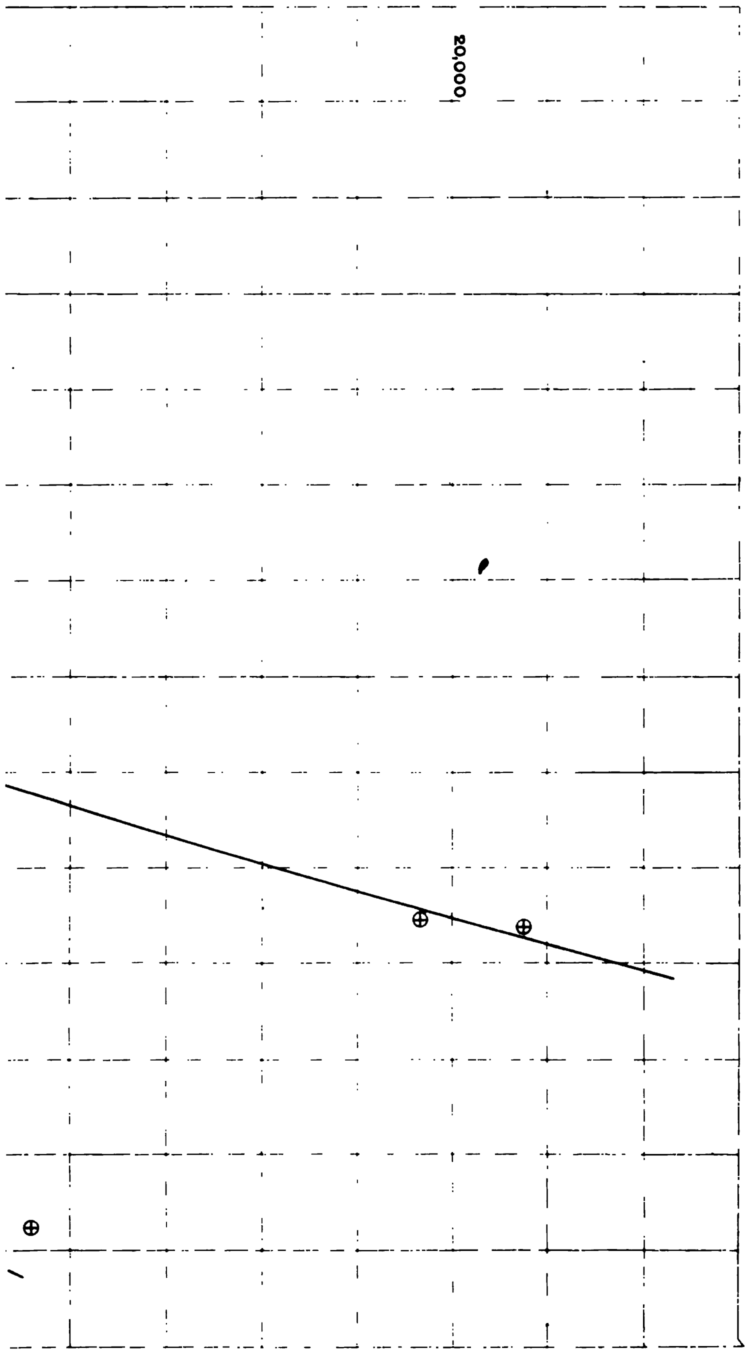

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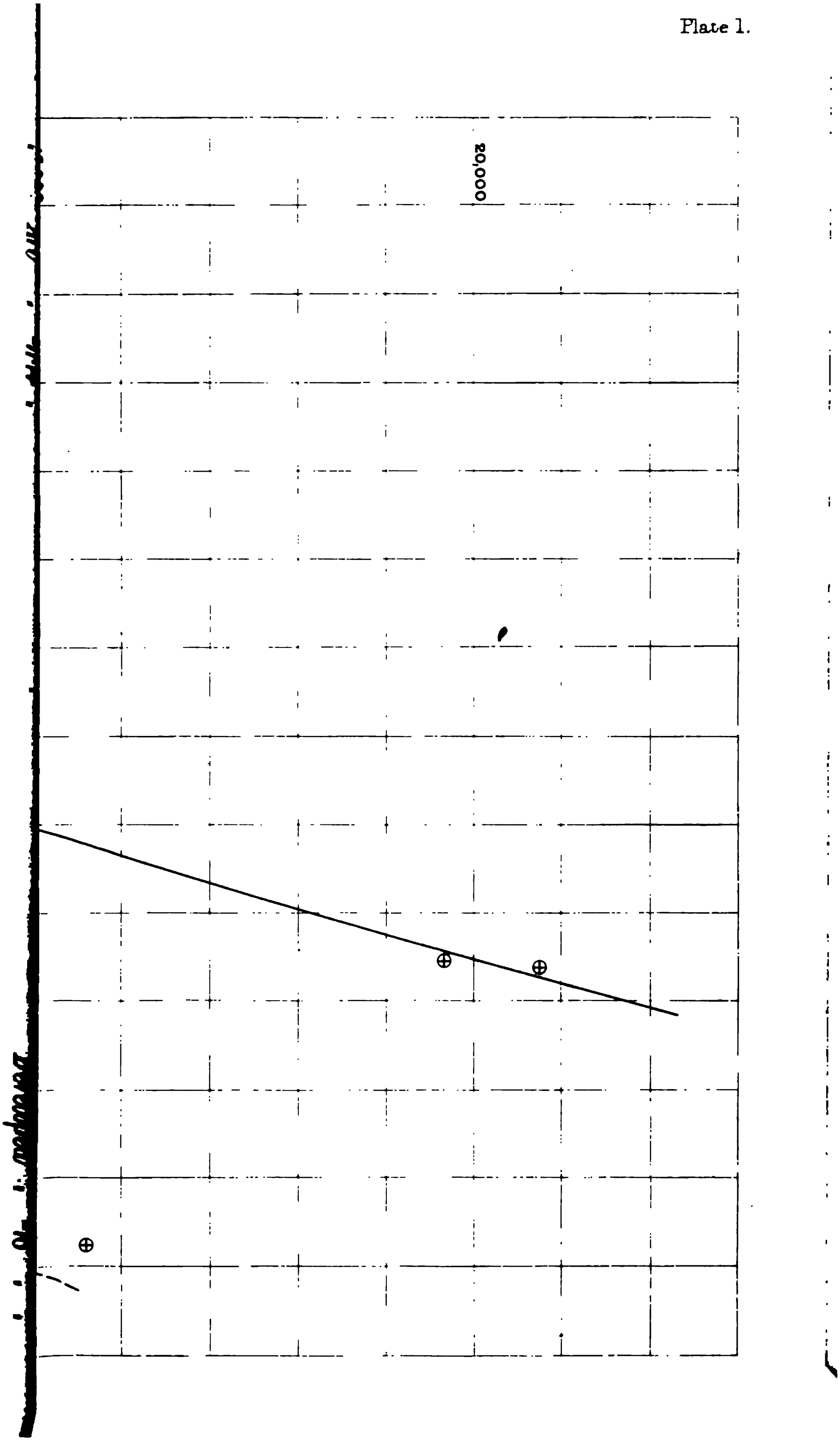

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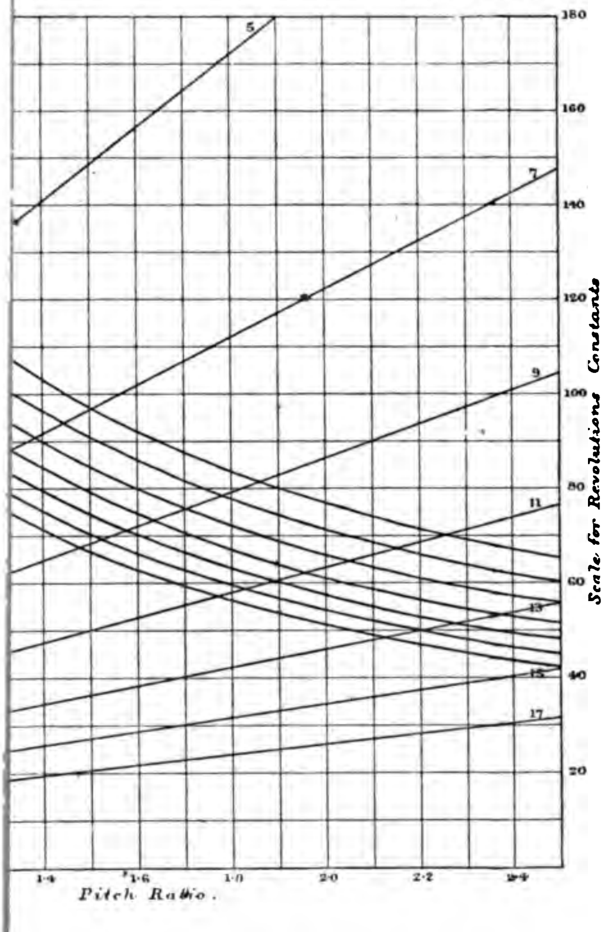
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Scale for Revolutions Constants
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1·8
2·0
2·2
2·4
Pitch Ratio.

Plate 3.

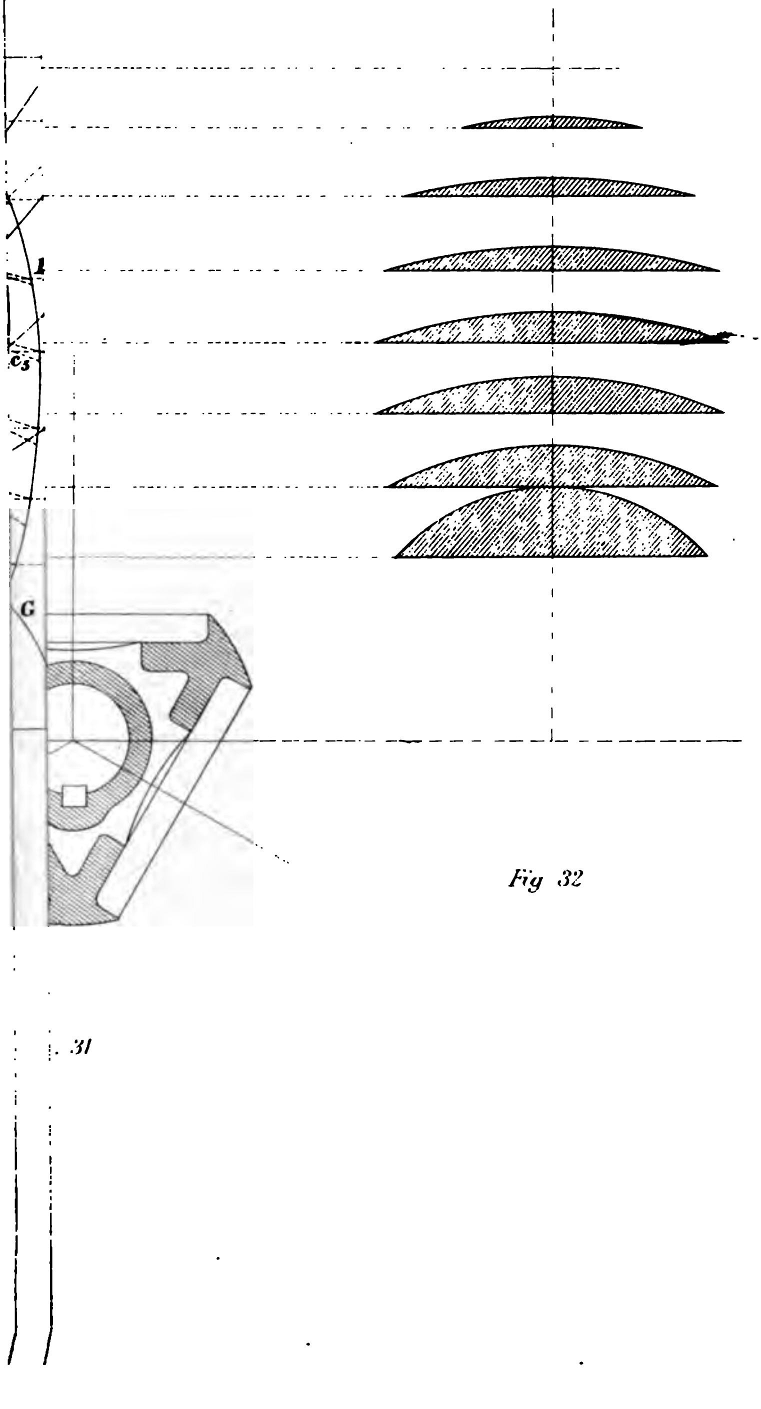

Fig 32

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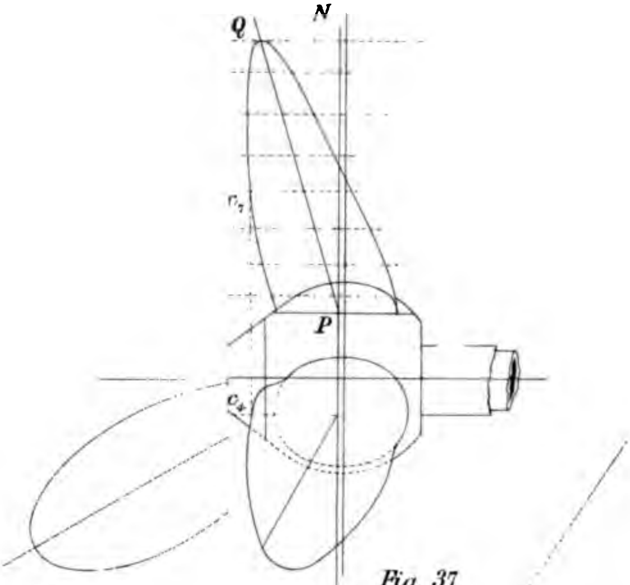


Fig. 37

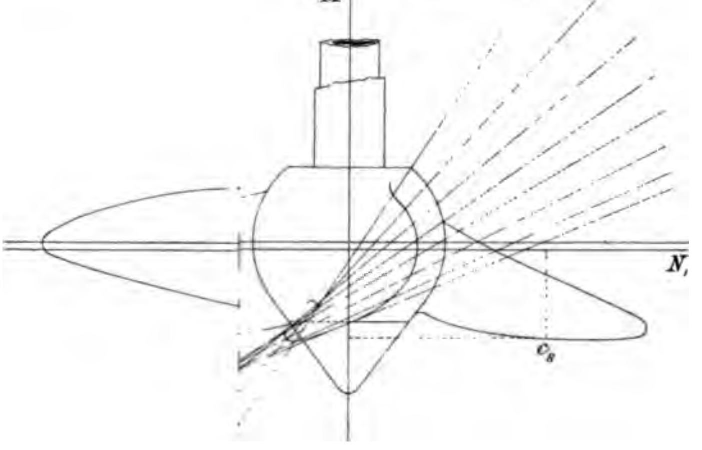
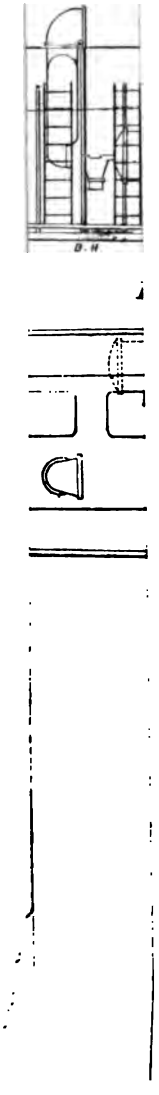
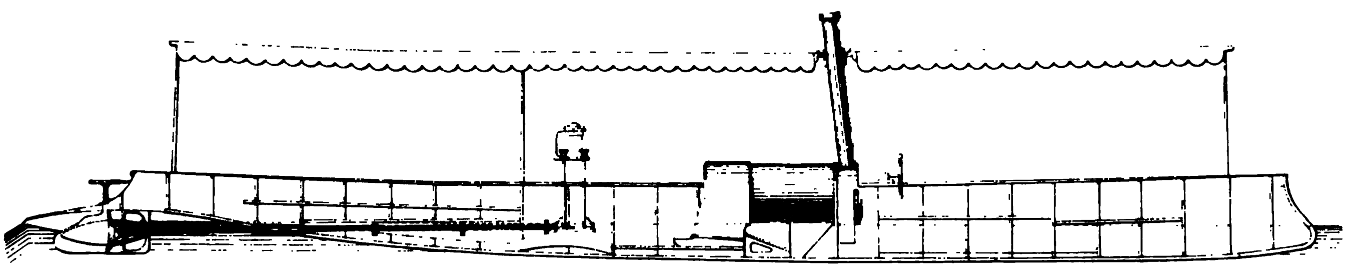


Fig. 38



STEAM LAUNCH FITTED WITH GUIDE-BLADE OR SCREW TURBINE PROPELLER.

Length	*45.0*
Beam	*6.0*
Draft	*1.0*
Speed	*13 Knots*



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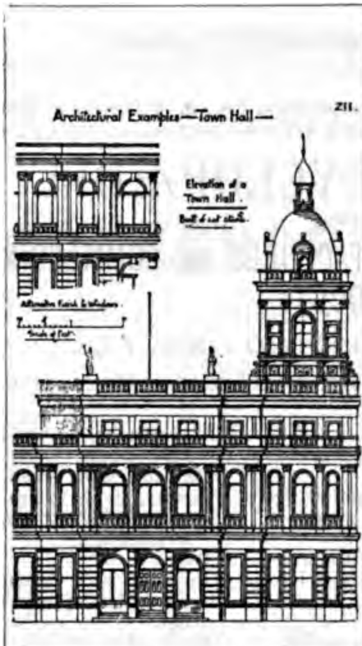

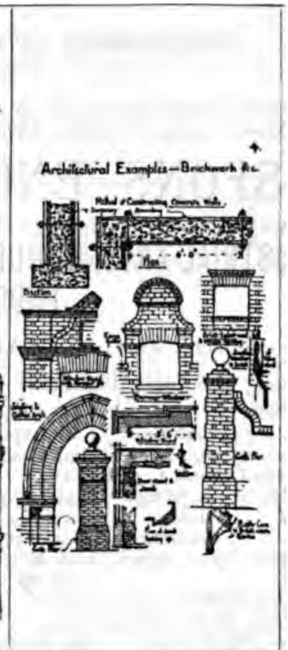

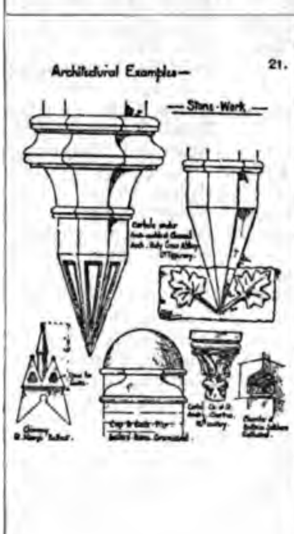

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